10/690666

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L15: Entry 2 of 3 File: USPT Aug 5, 2003

US-PAT-NO: 6602161
DOCUMENT-IDENTIFIER: US 6602161 B2

TITLE: Arrangement for operating the clutch in the power train of a motor vehicle

DATE-ISSUED: August 5, 2003

INVENTOR-INFORMATION:

NAME	CITY	STATE	ZIP CODE	COUNTRY
Hemmingsen; Fred Roar	Kongsberg			NO
Ring; Karl Frode	Asker			NO
Waerp; Ole Jonny	Kongsberg			NO
Sundet; Torgeir	Kongsberg			NO
Gunnerud; Morten Berger	Lurdalen			NO

ASSIGNEE-INFORMATION:

NAME	CITY	STATE	ZIP CODE	COUNTRY	TYPE CODE
LuK Lamellen und Kupplungsbau Beteiligungs KG	Buhl/Baden			DE	03

APPL-NO: 09/ 953231 [PALM]
DATE FILED: September 14, 2001

PARENT-CASE:
This is a continuation of international application Serial No. PCT/DE00/00800, filed Mar. 13, 2000, the entire disclosure of which is hereby incorporated by reference.

FOREIGN-APPL-PRIORITY-DATA:

COUNTRY	APPL-NO	APPL-DATE
DE	199 11 332	March 15, 1999

INT-CL: [07] B60 K 41/02, B60 K 41/24, B60 K 41/28

US-CL-ISSUED: 477/79; 192/3.55, 192/3.57, 477/80, 477/180, 701/67
US-CL-CURRENT: 477/79; 192/3.55, 192/3.57, 477/180, 477/80, 701/67

FIELD-OF-SEARCH: 477/79, 477/80, 477/176, 477/180, 192/3.55, 192/3.62, 192/3.57, 701/67

PRIOR-ART-DISCLOSED:

U.S. PATENT DOCUMENTS

Search Selected Search ALL Clear

	PAT-NO	ISSUE-DATE	PATENTEE-NAME	US-CL
☐	4576062	March 1986	Reppert et al.	192/222
☐	4825993	May 1989	Kurihara et al.	477/79
☐	5002170	March 1991	Parsons et al.	477/174
☐	5176234	January 1993	Reik et al.	192/52
☐	5337868	August 1994	Liu et al.	477/74
☐	5450934	September 1995	Maucher	192/70.25
☐	5626534	May 1997	Ashley et al.	477/79
☐	5819585	October 1998	Darnell	477/79
☐	5928110	July 1999	Vornehm et al.	477/114
☐	5993350	November 1999	Lawrie et al.	180/65.2
☐	6358186	March 2002	Kosik et al.	477/176

FOREIGN PATENT DOCUMENTS

FOREIGN-PAT-NO	PUBN-DATE	COUNTRY	US-CL
43 26 862 A 1	March 1994	DE	
0038 113	October 1981	EP	
0038 113	July 1986	EP	
0 601 728	June 1994	EP	
0696 341	February 1996	EP	
0 735 957	September 1998	EP	
WO 95/22013	August 1995	WO	
WO 97/03497	January 1997	WO	

ART-UNIT: 3681

PRIMARY-EXAMINER: Estremsky; Sherry

ASSISTANT-EXAMINER: Lewis; Tisha D.

ATTY-AGENT-FIRM: Darby & Darby

ABSTRACT:

The power train of a motor vehicle employs an electronic control unit which selects the extent of engagement of the friction clutch between the engine and the transmission by way of an electronic or fluid-operated actuator. The throttle valve for the engine is adjustable by the control unit and/or by the accelerator pedal to ensure predictable starting of the motor vehicle from standstill on a road surface having a pronounced slope and/or when the motor vehicle carries a heavy load.

21 Claims, 5 Drawing figures

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1666

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L15: Entry 3 of 3 File: USPT Mar 12, 1985

US-PAT-NO: 4503734
DOCUMENT-IDENTIFIER: US 4503734 A

TITLE: Control system for vehicular friction-type transmission clutch

DATE-ISSUED: March 12, 1985

INVENTOR-INFORMATION:

NAME	CITY	STATE	ZIP CODE	COUNTRY
Acker; Bernd	Esslingen			DE

ASSIGNEE-INFORMATION:

NAME	CITY	STATE	ZIP CODE	COUNTRY	TYPE CODE
Daimler-Benz Aktiengesellschaft				DE	03

APPL-NO: 06/ 402235 [PALM]
DATE FILED: July 27, 1982

FOREIGN-APPL-PRIORITY-DATA:

COUNTRY	APPL-NO	APPL-DATE
DE	3129681	July 28, 1981

INT-CL: [03] B60K 41/22

US-CL-ISSUED: 74/866; 192/3.58, 192/103R, 192/.032
US-CL-CURRENT: 701/67; 192/103R, 192/3.58, 477/154, 477/155

FIELD-OF-SEARCH: 192/3.58, 192/3.57, 192/.032, 192/.033, 192/.034, 192/.076, 192/.075, 192/13R, 192/19F, 192/.052, 192/.092, 74/866, 74/752A, 74/336R, 74/365

PRIOR-ART-DISCLOSED:

U.S. PATENT DOCUMENTS

Search Selected Search ALL Clear

PAT-NO	ISSUE-DATE	PATENTEE-NAME	US-CL
3898893	August 1975	Hashimoto et al.	74/866 X
3942393	March 1976	Forster et al.	74/752A
3956947	May 1976	Leising et al.	74/866
4106368	August 1978	Ivey	74/866

ART-UNIT: 352

PRIMARY-EXAMINER: Krizmanich; George H.

ATTY-AGENT-FIRM: Craig & Burns

ABSTRACT:

An arrangement for regulating the torque transmitted by friction type gear shifting elements whereby, at every point in time throughout a previously specified time duration of a gear shifting operation, the clutch torque which is necessary at that point in time and is defined as a minimum torque, is computed utilizing measured values of an input and output rotational speed of a change-speed transmission or a gearbox, as well as measured values of an operating pressure of the friction-type gear shifting elements. At the minimum torque the relative rotational speed of the parts of the friction-type gear shifting elements, for example, parts of a friction-type clutch, a difference between an input and output speed of the change-speed transmission or gearbox remains constant. At the same time, an additional torque is computed in accordance with a prespecified function. The operating pressure of the friction-type gear shifting elements is then adjusted analogously to an output signal U, with the output signal being an analog of a sum of the necessary torque and additional torque.

5 Claims, 2 Drawing figures

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L15: Entry 3 of 3 File: USPT Mar 12, 1985

DOCUMENT-IDENTIFIER: US 4503734 A
TITLE: Control system for vehicular friction-type transmission clutch

Abstract Text (1):
An arrangement for regulating the torque transmitted by friction type gear shifting elements whereby, at every point in time throughout a previously specified time duration of a gear shifting operation, the clutch torque which is necessary at that point in time and is defined as a minimum torque, is computed utilizing measured values of an input and output rotational speed of a change-speed transmission or a gearbox, as well as measured values of an operating pressure of the friction-type gear shifting elements. At the minimum torque the relative rotational speed of the parts of the friction-type gear shifting elements, for example, parts of a friction-type clutch, a difference between an input and output speed of the change-speed transmission or gearbox remains constant. At the same time, an additional torque is computed in accordance with a prespecified function. The operating pressure of the friction-type gear shifting elements is then adjusted analogously to an output signal U, with the output signal being an analog of a sum of the necessary torque and additional torque.

Brief Summary Text (4):
The aim underlying the present invention essentially resides in providing a regulating arrangement for regulating torque transmission by friction-type gear shifting elements such as, for example, clutch plates of a change-speed transmission, which arrangement enables a pressure at which the clutch plates are pressed together to be controlled in such a manner that a relative rotational speed between the clutch plates is brought to zero within a preselectable or gear shifting time reguardless of all external influences.

Brief Summary Text (5):
In accordance with advantageous features of the present invention, a regulating arrangement of the aforementioned type is proposed wherein an additional measured value sensor means or transmitter is provided for determining the clutch pressure, with the additional sensor means being located on an input side of a control device. The control device is adapted to generate, within presetable gear shifting times, an output signal which satisfies the following relationship:

Brief Summary Text (9):
p(t)=the clutch pressure at a point in time t;

Brief Summary Text (11):
.mu.=the coefficient of friction of the friction elements, i.e. friction clutch;

Brief Summary Text (12):
.eta.(t)=the relative rotational speed of the clutch element at the point in time t;

Brief Summary Text (15):
The present invention is based upon the premise or knowledge that a torque M.sub.C

transmitted by a friction element such as a friction clutch is additively made up from a torque M.sub.CN, necessary at the moment in question, and from an additional M.sub.CE, with M.sub.CN representing the minimum clutch torque which is necessary in order to maintain an instantaneous relative rotational speed of the clutch plates at a constant value, with a change in the relative rotational speeds of the clutch plates, as a function of time, being determined solely by the excess torque M.sub.CE.

Brief Summary Text (16):
It is thus generally true that the relative rotational speed of the clutch plates of the friction clutch increases when the torque M.sub.C, transmitted by the friction clutch, is less than the necessary torque M.sub.CN, and decreases when the transmitted torque M.sub.C is greater than the necessary torque M.sub.CN. If M.sub.C =M.sub.CN, the relative rotational speed, that is, the difference between the respective speeds of the clutch plates of the friction clutch, remains constant.

Brief Summary Text (17):
Therefore, if the necessary torque M.sub.CN, which incidentially corresponds to an average torque transmitted by the clutch in a closed or engaged state, can be determined sufficiently accurately, it is then possible by specifying an excess torque M.sub.CE, to predetermine the relative rotational speed of the clutch plates within a gear shifting time T.

Brief Summary Text (18):
The torque transmitted by the clutch may take the following relationship:

Brief Summary Text (20):
In accordance with the present invention, since M.sub.CN =M.sub.C -M.sub.CE, the necessary torque is consequently found for each point in time during a gear shifting operation, which is concluded within the gear shifting time T from a measured clutch pressure p, and from a variation, as a function of time, of the relative rotational speed of the clutch plates of the friction clutch which, in a straightforward manner, may be determined by measuring or sensing input and output speeds of the change-speed transmission or gear box.

Brief Summary Text (21):
The general concept underlying the present invention is a determination of a value of a torque M.sub.CN which is necessary at any particular given instant and is superimposed on the torque value and excess torque value. The torque value M.sub.CN is determined by measuring or sensing the input and output speeds of the change-speed transmission or gear box together with the clutch pressure while the excess torque M.sub.CE is determined from a predetermined function f(t) and from a measured or sensed relative rotational speed of the clutch plates of the friction clutch. Since a variation of the relative rotational speed between the clutch plates, as a function of time, is determined solely by the excess torque M.sub.CE, and since, the necessary torque M.sub.CN may readily be determined by measurements at any point in time, it is possible in accordance with advantageous features of the present invention to achieve a precisely predetermined gear shifting behavior accompanied by a gear shifting time which may be specified in advance and it is merely necessary that the function f.sub.t satisfy the above noted relationship.

Brief Summary Text (26):
It is possible, for example, with regard to comfort in shifting or the like to use constants e.sub.2 and j which may be freely specified in advance to define a maximum rate of change of the additional torque M.sub.CE, that is, the second differential, with respect to time, .eta.(t) of the relative rotational speed of the clutch plates at the beginning of the gear shifting operation by virtue of the following relationship: ##EQU3##

Brief Summary Text (28):
In accordance with the present invention, it may also be ensured that the relative rotational speed .eta. decreases or falls once the gear shifting time T has elaspsed to approximately zero without any jerk or uneven operation even when, for comfort related reasons, it is necessary to limit the torque M.sub.C transmitted by the friction elements such as a friction clutch or to limit a variation as a function of time, or when the dynamic characteristics of the actuation elements which are controlled by electronic control means do not permit the torques to be varied in the manner desired by the control unit.

Brief Summary Text (29):
The constants a.sub.1, a.sub.2 are predetermined by the construction of the change-speed transmission or gear box and, in some cases, by an overall design consideration of the vehicle in which the change-speed transmission or gear box is installed. A determination of the values for the constants a.sub.1, a.sub.2 may be determined in advance by the following relationship: ##EQU5## Wherein: c.sub.1 =the total moment of inertia of the engine and drive-line disposed forwardly of the change-speed transmission or gear box;

Brief Summary Text (30):
c.sub.2 =denotes the total moment of inertia of the drive-line following the gearbox or change-speed transmission, including the differential and vehicle body;

Brief Summary Text (34):
i.sub.R =the number of friction surfaces or number of plates in the friction element such as a friction clutch;

Brief Summary Text (35):
R.sub.m =an equavalent frictional radius of the plates of the friction clutch; and

Brief Summary Text (36):
F=a cross section of a cylinder-piston unit 6 for applying pressure to friction surfaces and/or friction plates of the friction element.

Brief Summary Text (42):
In performing the above calculation, it is particularly advantageous when the values for a.sub.1 and a.sub.2, which may have been incorrectly specified in advance, are compensated to a large extent by appropriately altering the calculated coefficient of friction .mu.. It is thus sufficient to specify mean or average, constant values for a.sub.1, and a.sub.2, for example, for a mean or average vehicle weight, without thereby significantly altering the gear shifting behavior when the vehicle characteristics change due to, for example, load changes of the vehicle.

Drawing Description Text (2):
FIG. 1 is a partially schematic cross sectional view through a multi-plate friction clutch adapted to be controlled by a regulating arrangement constructed in accordance with the present invention; and

Detailed Description Text (2):
Referring now to the drawings wherein like reference numerals are used in both views to designate like parts and, more particularly, to FIG. 1, according to this figure, a multi-plate clutch, of conventional construction, is rotationally symmetrically disposed with respect to an axis of rotation 1, with the clutch including an outer plate carrier 2 with outer plates located therein in such a manner so as to permit movement, and an inner plate carrier 4 provided with inner plates 5. A piston-cylinder unit generally designated by the reference numeral 6 is located on the outer plate carrier 2, with the unit 6 having an annular cross sectional configuration. An operating fluid such as, for example, oil, or the like, is fed through a bore 9 into an annular cylinder 10 of the cylinder-piston unit 6,

whereby an annular piston 7 of the cylinder-piston unit 6 may be moved against a biasing pressure or force of a spring 8. During a movement of the annular piston 7, the piston 7 presses the outer plates 3 against the inner plates 5. The annular cylinder 10 has an outer radius R.sub.a and an inner radius r.sub.a, with the inner plates 5 having an outer radius R.sub.p and an inner radius r.sub.p. The equivalent frictional radius R.sub.m of the plates may be determined in a conventional manner by the following relationship: ##EQU7##

Detailed Description Text (3):
As shown in FIG. 2, the system which is to be controlled by the regulating arrangement of the present invention includes a drive means 20 such as an engine or the like, having connected thereto a change-speed transmission or gearbox 21 which includes a clutch 22, with the drive 20 being adapted to propel or drive a vehicle 23. A valve 24 is provided for adjusting the pressure p of the clutch 22, with the valve 24 being controlled by an electronic control means 25, which is adapted to provide appropriate output control signals U to a control device (not shown) with an actuating element of the control device effecting an adjustment of the valve 24.

Detailed Description Text (4):
The control unit 25 is coupled, on an input side thereof, to measured value sensors or transmitters 26, 27, with the sensor 26 adapted to sense or measure a rotational speed .omega..sub.In on an input side of the change-speed transmission or gearbox 21, and the other measured value sensor 27 determining a rotational speed .omega..sub.Out on an output side of the change-speed transmission of gearbox 21. An additional measured value sensor or transmitter 28 is, according to the present invention, provided for determining a clutch pressure, that is, a pressure in the piston-cylinder unit 6 with the sensor 28 transmitting a measured value signal p' as an additional input to the control unit 25.

Detailed Description Text (13):
It is possible, in accordance with the present invention, to utilize a microprocessor which is available in the vehicle for other purposes such as, for example, controlling the ignition or braking system of the vehicle, to function as the control unit 25.

Current US Original Classification (1):
701/67

CLAIMS:

1. An arrangement for regulating a torque transmitted by friction-type gear-shifting means in a change-speed transmission means, wherein a gear shifting is accomplished under a load, the gear shifting means including a multiplate friction clutch means, the arrangement comprising a first sensor means for sensing an input speed of the transmission means and for providing an output signal of a sensed input speed, a second sensor means for sensing an output signal of the transmission means and for providing an output signal of the sensed output speed, characterized in that a third sensor means is arranged at the transmission means for sensing an operating pressure of an operating fluid of the friction clutch means and for providing an output signal of a sensed operating pressure, and in that a control means is provided for receiving, as an input, the output signals of the first, second, and third sensor means and for generating, within a presetable gear shifting time, an output signal for the operating pressure of the friction clutch means which satisfies the following relationship: for o<t<T,

U(t)=a.sub.2 .mu.p(t)-1/a.sub.1 .eta.(t)+f(t).eta.(t),

wherein:

T=a duration of the gear shifting operation, that is, a gear-shifting time,

t=a point in time during the gear-shifting operation, that is, 0.ltoreq.t.ltoreq.T;

p(t)=the operating pressure at the point in time t;

a.sub.1, a.sub.2 =constants specified in advance;

.mu.=a coefficient of friction of the friction clutch means;

.eta.(t)=a relative rotational speed between plates of the multiplate friction clutch means at the point in time t;

.eta.(t)=a differential, with respect to time, of a relative rotational speed at a point in time t; and

f(t)=a continuous function, specified in advance, satisfying the following relationship:

f(0).eta.(0)=f(T).eta.(T)=0.

4. An arrangement according to one of claims 1, 2, or 3, for a motor vehicle, characterized in that values for the constants a.sub.1, a.sub.2 are determined in accordance with the following relationship: ##EQU9## wherein: c.sub.1 =a total moment of inertia of the drive means of the vehicle and a drive line means located forwardly of the change-speed transmission means;

c.sub.2 =a total moment of inertia of a drive-line located rearwardly of the change-speed transmission means, with the rearwardly located drive line including a differential and a body portion of the vehicle;

c.sub.3 =an inertial mass of coupling shafts connected to a planetary gear means of the differential, which shafts rotate with a remainder of the system;

.gamma.=a servo-moment factor;

i=a transmission ratio of a gear of the change-speed transmission means to be engaged;

i.sub.R =a number of one of friction-surfaces and clutch plates in the friction clutch means;

R.sub.m =an equivalent frictional radius of the clutch plates of the friction clutch means; and

F=a cross section of a cylinder-piston means for applying the operating pressure to one of the friction surfaces and clutch plates of the friction clutch means.

5. An arrangement according to claim 4, characterized in that the coefficient of friction .mu. of the friction clutch means is determined in accordance with the following relationship: ##EQU10## wherein:

.ltoreq. t.sub.1 <t.sub.2 .ltoreq.T,

with t.sub.1 and t.sub.2 being two successive time intervals determined by the control means.

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L11: Entry 1 of 1 File: USPT May 24, 2005

DOCUMENT-IDENTIFIER: US 6898504 B2
TITLE: Vehicle driving force control apparatus

Detailed Description Text (78):
The value GDV in the previous equation is a constant that serves to convert the average rate of change DNsr into a value corresponding to the response delay of the clutch operation (i.e., a constant that serves to convert the average rate of change into the amount by which the converted output shaft rotational speed Nsr increases or decreases during the response delay period of the clutch 12). In other words, due to the response delay of the clutch operation, it is anticipated that the larger the average rate of change DNsr is, the larger will be the amount by which the rotational speed of the clutch output shaft exceeds the aforementioned detected value when the clutch 12 actually connects. This amount is corrected with the value DNsr.times.GDV.

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L6: Entry 1 of 1 File: USPT Feb 26, 2002

US-PAT-NO: 6351700
DOCUMENT-IDENTIFIER: US 6351700 B1

TITLE: Speed change controller and control method of infinite speed ratio continuously variable transmission

DATE-ISSUED: February 26, 2002

INVENTOR-INFORMATION:

NAME	CITY	STATE	ZIP CODE	COUNTRY
Muramoto; Itsuro	Kanagawa			JP
Kawabe; Taketoshi	Kanagawa			JP
Joe; Shin-ichiro	Kanagawa			JP
Narita; Yasushi	Kanagawa			JP
Sakai; Hiromasa	Kanagawa			JP
Nishio; Motoharu	Kanagawa			JP

ASSIGNEE-INFORMATION:

NAME	CITY	STATE	ZIP CODE	COUNTRY	TYPE CODE
Nissan Motor Co., Ltd.	Yokohama			JP	03

APPL-NO: 09/ 605757 [PALM]
DATE FILED: June 28, 2000

FOREIGN-APPL-PRIORITY-DATA:

COUNTRY	APPL-NO	APPL-DATE
JP	11-181842	June 28, 1999
JP	12-138161	May 11, 2000

INT-CL: [07] G06 F 7/00

US-CL-ISSUED: 701/51; 701/67, 477/39
US-CL-CURRENT: 701/51; 477/39, 701/67

FIELD-OF-SEARCH: 701/51, 701/55, 701/56, 701/67, 477/31, 477/37, 477/39

PRIOR-ART-DISCLOSED:

U.S. PATENT DOCUMENTS

Search Selected Search ALL Clear

	PAT-NO	ISSUE-DATE	PATENTEE-NAME	US-CL
☐	5288281	February 1994	Perry	475/191
☐	5564998	October 1996	Fellows	475/216
☐	5766105	June 1998	Fellows et al.	474/18
☐	5885185	March 1999	Kidokoro et al.	476/10

FOREIGN PATENT DOCUMENTS

FOREIGN-PAT-NO	PUBN-DATE	COUNTRY	US-CL
9-89071	March 1997	JP	
10-267116	October 1998	JP	

ART-UNIT: 3661

PRIMARY-EXAMINER: Beaulieu; Yonel

ATTY-AGENT-FIRM: Foley & Lardner

ABSTRACT:

An infinite speed ratio continuously variable transmission is provided with a continuously variable transmission (2), fixed speed ratio transmission (3), planetary gear set (5), power recirculation clutch (9) and direct clutch (10). A target speed ratio of the infinite speed ratio continuously variable transmission is set based on a vehicle speed and an accelerator pedal depression amount. When the target speed ratio varies beyond a rotation synchronous point, a control unit (80) assigns an order of priority to control of the power recirculation clutch (9) and direct clutch (10), and control of the speed ratio of the continuously variable transmission (2), and thereby causes a real speed ratio of the infinite speed ratio continuously variable transmission to vary in the same direction until it reaches the target speed ratio (S21, S22, S31, S32, S121, S122).

11 Claims, 18 Drawing figures

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701/67

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Searching 1976 to present:

Results of Search in 1976 to present db for:
(((SPEC/"automatic clutch control" AND SPEC/speed) AND SPEC/friction) AND SPEC/coefficient): 8 patents.

1 5,847,272 T Function testing device for an actuator system such as a automatic friction clutch used with a motor vehicle gearbox
2 5,590,563 T Electronically controlled transmission
3 5,439,428 T Method and apparatus for robust automatic clutch control with pid regulation
4 5,403,249 T Method and apparatus for robust automatic clutch control
5 5,277,286 T Method of controlling automatic clutch for motor vehicles
6 4,778,038 T Control apparatus for automobile clutch
7 4,632,231 T Method of controlling the starting of a vehicle having automatic clutch
8 4,618,043 T Method for automatic control of a motor vehicle clutch

Searching 1976 to present...

Results of Search in 1976 to present db for:
(((SPEC/"automatic clutch control" AND SPEC/speed) AND SPEC/transmission) AND SPEC/coefficient): 9 patents.

1 6,769,526 T Apparatus for estimating clutch temperature
2 ~~5,847,272~~ T Function testing device for an actuator system such as a automatic friction clutch used with a motor vehicle gearbox
3 5,590,563 T Electronically controlled transmission
4 5,439,428 T Method and apparatus for robust automatic clutch control with pid regulation
5 5,403,249 T Method and apparatus for robust automatic clutch

control

6 5,277,286 T Method of controlling automatic clutch for motor vehicles

7 4,778,038 T Control apparatus for automobile clutch

8 4,632,231 T Method of controlling the starting of a vehicle having automatic clutch

9 4,618,043 T Method for automatic control of a motor vehicle clutch

L6: Entry 1 of 1 File: USPT Feb 26, 2002

DOCUMENT-IDENTIFIER: US 6351700 B1
TITLE: Speed change controller and control method of infinite speed ratio continuously variable transmission

Abstract Text (1):
An infinite speed ratio continuously variable transmission is provided with a continuously variable transmission (2), fixed speed ratio transmission (3), planetary gear set (5), power recirculation clutch (9) and direct clutch (10). A target speed ratio of the infinite speed ratio continuously variable transmission is set based on a vehicle speed and an accelerator pedal depression amount. When the target speed ratio varies beyond a rotation synchronous point, a control unit (80) assigns an order of priority to control of the power recirculation clutch (9) and direct clutch (10), and control of the speed ratio of the continuously variable transmission (2), and thereby causes a real speed ratio of the infinite speed ratio continuously variable transmission to vary in the same direction until it reaches the target speed ratio (S21, S22, S31, S32, S121, S122).

Brief Summary Text (2):
This invention relates to control of an infinite speed ratio continuously variable transmission for vehicles.

Brief Summary Text (4):
Tokkai Hei10-267116 published by the Japanese Patent Office in 1998 discloses an infinite speed ratio continuously variable transmission for a vehicle (referred to hereafter as IVT) which can vary a speed ratio continuously to infinity by combining a reduction gear and a planetary gear set with a toroidal continuously variable transmission (referred to hereafter as CVT).

Brief Summary Text (5):
The rotation of an engine is input to the CVT whereof the speed ratio can be varied continuously, and a reduction gear having a fixed speed change ratio. The rotation of an output shaft of the CVT is input to a sun gear of the planetary gear set, and the rotation of an output shaft of the reduction gear is input to a planet carrier of the planetary gear set. The final output shaft of the IVT is joined to a ring gear of the planetary gear set. The output shaft of the reduction gear and planet carrier are joined via a power recirculation clutch. The output shaft of the CVT is also joined to the final output shaft via a direct clutch.

Brief Summary Text (6):
In a power recirculation mode wherein the power recirculation clutch is engaged and the direct clutch is disengaged, the direction and speed of output rotation of the final output shaft varies according to the difference of the speed ratio of the CVT, and the speed ratio of the reduction gear. Specifically, the speed ratio Ii of the IVT, i.e., the value of the input shaft rotation speed/final output shaft rotation speed of the IVT varies from a negative value to a positive value. Also, at the point where the rotation direction of the final output shaft varies, the speed ratio of the IVT Ii is infinite, and the final output shaft is stationary. This point is referred to as a geared neutral point (GNP).

Brief Summary Text (7):
On the other hand, in a direct mode wherein the power recirculation clutch is disengaged and the direct clutch is engaged, the rotation of the CVT output shaft is output to the final output shaft without modification, so the speed ratio Ii of the IVT is equal to the CVT speed ratio IC.

Brief Summary Text (8):
The change-over between the power recirculation mode and the direct mode is performed at a rotation synchronous point (RSP) at which the IVT speed ratio Ii of each mode coincide with each other. If we define the inverse of the IVT speed ratio Ii as an IVT speed ratio factor .EPSILON., the IVT speed ratio factor .EPSILON. increases as the vehicle starts to move forward and accelerate in the power recirculation mode. The CVT speed ratio Ic also increases at that time. At the RSP, a mode change-over is performed, and in the subsequent direct mode, the IVT speed ratio factor .EPSILON. continues to increase but the CVT speed ratio IC decreases. Also, reversing of the vehicle is performed only in the power recirculation mode, wherein the IVT speed ratio factor .EPSILON. increases in negative value and the CVT speed ratio Ic approaches O together with the acceleration of the vehicle.

Brief Summary Text (10):
During a kickdown operation of the IVT, i.e., an increases of the speed ratio of the IVT due to the depression of an accelerator pedal of the vehicle by a driver, or during a shift-up operation of the IVT when the driver releases his foot from the accelerator pedal, a rapid speed change is required. If this speed change is performed beyond the rotation synchronous point RSP, the CVT speed ratio Ic is temporarily fixed at the rotation synchronous point RSP, and the engagement and disengagement of the power recirculation clutch and direct clutch are changed over. However, this change-over delays variation of the IVT speed ratio Ii.

Brief Summary Text (11):
One way of enabling rapid speed change would be to make the change-over between the power recirculation mode and direct mode while the clutches are partially engaged. However, the change-over of the operation mode of the IVT in the course of IVT speed ratio control towards a target value may cause the IVT speed ratio to overrun the target value. In this case, the shift direction of the IVT abruptly reverses immediately after the change-over of operation mode so as to cancel out the overrun, and the driver may experience an uncomfortable feeling due to the abrupt change of the shift direction of the IVT.

Brief Summary Text (13):
In order to achieve the above object, this invention provides a speed change controller for such an infinite speed ratio continuously variable transmission for use with a vehicle that comprises an input shaft, a continuously variable transmission which transmits a rotation speed of the input shaft to a continuously variable transmission output shaft at an arbitrary speed ratio, a fixed speed ratio transmission which transmits the rotation speed of the input shaft to a fixed speed ratio transmission output shaft at a fixed speed ratio, a planetary gear set comprising a first rotation member joined to the continuously variable transmission output shaft, a second rotation member joined to the fixed speed ratio transmission output shaft, and a third rotation member which varies a rotation direction and a rotation speed according to a difference between a rotation speed of the first rotation member and a rotation speed of the second rotation member, a direct clutch which connects and disconnects the continuously variable transmission output shaft and the third rotation member, and a power recirculation clutch which connects and disconnects the fixed speed ratio transmission output shaft and the second rotation member.

Brief Summary Text (14):
The speed change controller comprises a sensor which detects a running state of the

vehicle, a sensor which detects a real speed ratio of the infinite speed ratio continuously variable transmission, and a microprocessor programmed to set a target speed ratio of the infinite speed ratio continuously variable transmission based on the running state, determine whether or not the target speed ratio has varied beyond a predetermined speed ratio, determine a priority of controlling the power recirculation clutch and direct clutch, and controlling the speed ratio of the continuously variable transmission, so that, when the target speed ratio has varied beyond the predetermined speed ratio, the real speed ratio of the infinite speed ratio continuously variable transmission varies in the same direction until it reaches the target speed ratio, and control the power recirculation clutch and direct clutch, and control the speed ratio of the continuously variable transmission, according to the priority.

Brief Summary Text (15):
This invention also provides a speed change controller comprising a mechanism for detecting a running state of the vehicle, a mechanism for detecting a real speed ratio of the infinite speed ratio continuously variable transmission, a mechanism for setting a target speed ratio of the infinite speed ratio continuously variable transmission based on the running state, a mechanism for determining whether or not the target speed ratio has varied beyond a predetermined speed ratio, a mechanism for determining a priority of controlling the power recirculation clutch and direct clutch, and controlling the speed ratio of the continuously variable transmission, so that, when the target speed ratio has varied beyond the predetermined speed ratio, the real speed ratio of the infinite speed ratio continuously variable transmission varies in the same direction until the real speed ratio reaches the target speed ratio, and a mechanism for performing control of the power recirculation clutch and direct clutch, and of the speed ratio of the continuously variable transmission, according to the priority.

Brief Summary Text (16):
This invention also provides a speed change control method for the infinite speed ratio continuously variable transmission for use with a vehicle. The method comprises detecting a running state of the vehicle, detecting a real speed ratio of the infinite speed ratio continuously variable transmission, setting a target speed ratio of the infinite speed ratio continuously variable transmission based on the running state, determining whether or not the target speed ratio has varied beyond a predetermined speed ratio, determining a priority of controlling the power recirculation clutch and direct clutch, and controlling the speed ratio of the continuously variable transmission, so that, when the target speed ratio has varied beyond the predetermined speed ratio, the real speed ratio of the infinite speed ratio continuously variable transmission varies in the same direction until the real speed ratio reaches the target speed ratio, and performing control of the power recirculation clutch and direct clutch, and of the speed ratio of the continuously variable transmission, according to the priority.

Detailed Description Text (2):
Referring to FIG. 1 of the drawings, an IVT for a vehicle comprises an input shaft 1, toroidal continuously variable transmission (CVT) 2, reduction gear 3, planetary gear set 5, final output shaft 6, power recirculation clutch 9 and direct clutch 10.

Detailed Description Text (4):
The CVT output shaft 4 is joined to a sun gear 5A of the planetary gear set 5, and is also joined to the final output shaft 6 via the direct clutch 10.

Detailed Description Text (5):
The reduction gear 3 comprises a gear 3A which is joined to the input shaft 1, a gear 3B and a gear output shaft 3C. The gear 3A engages with the gear 3B which is fixed to the gear output shaft 3C. The gear output shaft 3C is joined to a planet carrier 5B which carries plural planet gears 5D of the planetary gear set 5 via the

power recirculation clutch 9. A ring gear 5C of the planetary gear set 5 is joined to the final output shaft 6.

Detailed Description Text (6):
The rotation of the final output shaft 6 is transmitted to drive wheels 11A, 11B of the vehicle via a transmission output gear 7, final gear 12 and differential gear 8.

Detailed Description Text (7):
In this IVT, the drive wheels 11A, 11B are driven by one of two power transmission modes, i.e., a power recirculation mode wherein the power recirculation clutch 9 is engaged and the direct clutch 10 is disengaged, and a direct mode wherein the power recirculation clutch 9 is disengaged and the direct clutch 10 is engaged.

Detailed Description Text (9):
At the geared neutral point GNP, the ring gear 5C, i.e., the final output shaft 6, does not rotate, and the vehicle is at rest. When the CVT speed ratio Ic increases beyond the geared neutral point GNP, the ring gear 5C rotates in a forward direction, and when the CVT speed ratio Ic decreases to less than the GNP, the ring gear 5C rotates in a reverse direction. In other words, in the power recirculation mode, forward motion and reverse motion of the vehicle can be changed over by controlling the CVT speed ratio Ic.

Detailed Description Text (10):
When the vehicle is moving forward in the power recirculation mode, the IVT speed ratio Ii decreases as the CVT speed ratio Ic increases. In other words, the IVT speed ratio factor .EPSILON.=1/i increases as shown in FIG. 8.

Detailed Description Text (11):
When the IVT speed ratio factor E reaches the rotation synchronous point RSP, there is a shift from the power recirculation mode to the direct mode of the IVT. In the direct mode, the rotation of the CVT output shaft 4 is output to the final output shaft 6 directly, so the IVT speed ratio factor .EPSILON. increases as the CVT speed ratio Ic decreases. When the vehicle decelerates during travel, the CVT speed ratio Ic varies in an opposite direction to its variation during acceleration. This characteristic of the IVT of is disclosed in Tokkai Hei 9-89071 published by the Japanese Patent Office in 1997.

Detailed Description Text (13):
The control of the IVT speed ratio Ii comprises a control of the CVT speed ratio Ic, and the engaging and disengaging operation of the power recirculation clutch 9 and direct clutch 10. These controls are performed by the control unit 80 shown in FIG. 2. The control unit 80 comprises a central processing unit (CPU), read-only memory (ROM), random access memory (RAM) and input/output interface (I/O interface).

Detailed Description Text (14):
Signals showing detection values are input to the control unit 80 respectively from a rotation speed sensor 81 which detects a rotation speed Ni of the input shaft 1, rotation speed sensor 82 which detects the rotation speed No of the CVT output shaft 4, vehicle speed sensor 83 which detects a vehicle speed VSP from a rotation speed Nout of the final output shaft 6, inhibitor switch 85 which detects a selection position of a selector lever with which the vehicle is provided, and an accelerator pedal depression amount sensor 84 which detects an accelerator pedal depression amount APS of the vehicle. The input shaft rotation speed Ni is equal to the engine rotation speed Ne, therefore a crank angle sensor which detects the engine rotation speed Ne may be used instead of the rotation speed sensor 81. The vehicle speed VSP is obtained by multiplying the rotation speed Nout of the final output shaft 6 by a predetermined constant.

Detailed Description Text (15):
The control unit 80 determines a target IVT speed ratio TIiO based on these input signals, and controls the CVT speed ratio Ic to be equal to the target IVT speed ratio TIiO by a signal output to the step motor 36. The power recirculation clutch 9 and direct clutch 10 are selectively engaged via solenoids 91, 92 so as to change over between the power recirculation mode and direct mode.

Detailed Description Text (16):
The control of the power recirculation clutch 9 comprises the continuous variation of an engaging force by duty controlling the solenoid 91 so as to produce a partially engaged state. Likewise, the control of the direct clutch 10 comprises continuous variation of an engaging force by duty controlling the solenoid 92 so as to produce a partially engaged state.

Detailed Description Text (19):
In a step S1, the control unit 80 reads the input shaft rotation speed Ni, CVT output shaft rotation speed No, vehicle speed VSP and accelerator pedal depression amount APS detected by the above sensors.

Detailed Description Text (20):
In a step S2, a target input shaft rotation speed TNi is calculated from the accelerator pedal depression amount APS and vehicle speed VSP by looking up a map shown in FIG. 6. A final IVT speed ratio TIc is calculated by dividing the calculated target input shaft rotation speed TNi by the final output shaft rotation speed Nout detected by the vehicle speed sensor 83. Further, the target IVT speed ratio TIcO which takes account of a predetermined speed ratio response delay is calculated by processing the final IVT speed ratio TIc with a lowpass filter. It may be noted that the final IVT speed ratio TIc is a final target value of the IVT speed ratio Ii, and the target IVT speed ratio TIiO is a transient target value each time the routine controlling the IVT speed ratio Ii is executed, which is required for achieving the final IVT speed ratio TIi with a predetermined response speed. Further, from the map of FIG. 6, a running mode TMode is determined to achieve the final IVT speed ratio TIiO.

Detailed Description Text (21):
In the map of FIG. 6, when the target input shaft rotation speed TNi is in the region on the left of a mode change-over line shown by a dotted line, the power recirculation mode is the target running mode TMode, and when it is in the region on the right of the mode change-over line, the direct mode is the target running mode TMode.

Detailed Description Text (34):
In the step S13, a running mode change-over including shift-up of the CVT 2 is performed by a subroutine described later. This operation comprises changing over the clutches 9 and 10, and decreasing the CVT speed ratio Ic.

Detailed Description Text (35):
In the next step S14, it is determined whether the mode change-over operation including shift-up of the CVT 2 has been completed. This is done by determining whether or not a clutch change-over flag Fc and CVT speed ratio control flag FIc have both been reset to 0. If both of these processes have terminated, it is determined that operations have been completed, and the routine proceeds to a step S15.

Detailed Description Text (37):
On the other hand, if one of the clutch change over flag Fc and CVT speed ratio control flag FIc is 1, it signifies that the running mode change-over operation including shift-up of the CVT 2 is not complete, so the routine is terminated without modifying the shift-up flag Fup. Due to this processing, a running mode change-over operation including shift-up of the CVT 2 is performed again in the

step S13 on the next occasion that the main routine is performed.

Detailed Description Text (39):
In the step S17, a running mode change-over operation including shift-down of the CVT 2 is performed by a subroutine described later. This operation comprises change-over of the clutches 9 and 10, and increase of the CVT speed ratio Ic.

Detailed Description Text (40):
In a next step S18, it is determined whether or not a running mode change-over operation including shift-down of the CVT 2 has been completed. This is done by determining whether or not both the clutch change-over flag Fc and the CVT speed ratio control flag FIc have been reset to 0. When both of these processes have both terminated, it is determined that operations have been completed, and the routine proceeds to a step S19.

Detailed Description Text (42):
In the step S18, when it is determined that one of the clutch change-over flag Fc and CVT speed ratio control flag FIc is 1, running mode change-over control including shift-down of the CVT 2 is not complete, so the routine is terminated without modifying the shift-down flag Fdown. Due to this processing, a running mode change-over operation including shift-down of the CVT 2 is performed again in the step S17 on the next occasion that the main routine is performed.

Detailed Description Text (45):
In a first step S21, it is determined whether or not the clutch change-over flag Fc is 1. When the clutch change-over flag Fc is 1, the subroutine proceeds to a step S24. When the clutch change-over flag Fc is zero, the subroutine proceeds to a step S22.

Detailed Description Text (46):
In the step S22, it is determined whether or not the CVT speed ratio control flag FIc is set to 1. When the CVT speed ratio control flag FIc is 0, the clutch change-over flag Fc is set to 1 in a step S23, and the routine proceeds to the step S24.

Detailed Description Text (47):
In the step S24, clutch change-over is performed.

Detailed Description Text (48):
This operation is performed by controlling the excitation state of the solenoids 91, 92. When there is a shift from the power recirculation mode to the direct mode, a partially engaged state of the clutches 9, 10 is produced by gradually disengaging the power recirculation clutch 9 from the engaged state, and gradually engaging the direct clutch 10 from the disengaged state. After passing through this state, the direct clutch 10 is engaged and the power recirculation clutch 9 is disengaged. The change-over from the direct mode to the power recirculation mode is performed by the reverse process to the above.

Detailed Description Text (49):
In a next step S25, it is determined whether or not clutch change-over has been completed. As a result of clutch change-over, the IVT speed ratio Ii varies relative to the same CVT speed ratio Ic. Therefore, this determination may be performed by determining, for example at a predetermined CVT speed ratio Ica shown in FIG. 8, whether there has been a change from an IVT speed ratio Iic to Iia, or whether there has been a change in the reverse direction.

Detailed Description Text (50):
When clutch change-over is not complete in the step S25, the routine is terminated without performing other steps. As a result, clutch change-over is performed again in the step S24 on the next occasion this subroutine is executed. When clutch change-over is complete, the subroutine proceeds to a step S26.

Detailed Description Text (51):
In the step S26, the clutch change-over flag Fc is reset to 0, the CVT speed ratio control flag FIc is set to 1 in the following step S27, and the subroutine is terminated.

Detailed Description Text (56):
In this subroutine, firstly, the clutch change-over operation is completed, and a shift-up operation of the CVT 2 is then performed.

Detailed Description Text (57):
In FIG. 8, when there is a shift-down of the IVT from a point A to a point B, the control unit 80 first engages the power recirculation clutch 9 and disengages the direct clutch 10 while the CVT speed ratio Ic is fixed at Ica. As a result, the IVT speed ratio Ii increases from Iia to Iic. Subsequently, the CVT 2 shifts down from the speed ratio Ica to a speed ratio Icb due to the signal output to the step motor 36. Due to this variation of the CVT speed ratio Ic, the IVT speed ratio Ii increases further from Iic to Iib. In other words, the IVT speed ratio Ii shifts down progressively from the point A to the point B via the point C. Therefore, there is no change in the shift direction of the IVT during the operation, as shown in FIG. 9.

Detailed Description Text (58):
On the other hand, if the CVT speed ratio Ic varies from Ica to a value Icb corresponding to the point B and a clutch change-over is then performed, the IVT speed ratio Ii reaches the point B from the point A via a point D, so the shift direction does vary during the operation as shown by the dotted line in FIG. 9.

Detailed Description Text (59):
Next, in FIG. 8, the case will be considered where the IVT shifts up from the point C to the point D. In this case also, the control unit 80 first disengages the power recirculation clutch 9 and engages the direct clutch 10 while the CVT speed ratio Ic is fixed at Ica. As a result, the IVT speed ratio Ii decreases from Iic to Iia. Subsequently, the control unit 18 shifts the CVT speed ratio Ic up from Ica to Icb due to a signal output to the step motor 36. As a result of this variation of the CVT speed ratio Ic, the IVT speed ratio Ii further decreases from Iia to Iid. In other words, the IVT speed ratio Ii progressively shifts up from the point C to the point D via the point A. Therefore, in this case also there is no change in the shift direction of the IVT during the operation. On the other hand, if the CVT speed ratio Ic is first varied from Ica to Icb and a clutch change-over is then performed, the IVT speed ratio Ii reaches the point D from the point C via the point B, so there is a change in the shift direction of the IVT during the operation.

Detailed Description Text (62):
In the step S32, it is determined whether or not the clutch change-over flag Fc is 1. When the clutch change-over flag Fc is 0, the CVT speed ratio control flag FIc is set to 1 in a step S33, and the routine proceeds to a step S37.

Detailed Description Text (65):
In the step S39, the CVT speed ratio control flag FIc is reset to 0, and the routine proceeds to a step S40. In the step S40, the clutch change-over flag Fc is set to 1 and the subroutine is terminated.

Detailed Description Text (66):
On the other hand, in the step S34, clutch change-over is performed.

Detailed Description Text (67):
This operation is performed by controlling the excitation states of the solenoids 91, 92. When there is a shift from the direct mode to the power recirculation mode,

a partially engaged state of the clutches 9, 10 is produced by gradually disengaging the direct clutch 10 from the engaged state, and gradually engaging the power recirculation clutch 9 from the disengaged state. After passing through this state, the power recirculation clutch 9 is engaged and the direct clutch 10 is disengaged. The change-over from the direct mode to the power recirculation mode is performed by the reverse process to the above.

Detailed Description Text (68):
In a next step S35, is determined whether or not clutch change-over has been completed by an identical method to that of the step S25 of FIG. 4. When change-over is not complete, the subroutine is terminated without performing further steps. As a result, clutch change-over is again performed in the step S34 on the next occasion that this subroutine is performed. When it is determined that clutch change-over is complete, the subroutine proceeds to a step S36. In the step S36, the clutch change -over flag Fc is reset to 0 and the subroutine is terminated.

Detailed Description Text (69):
In this subroutine, the shift-down operation of the CVT 2 is first completed, and the clutch change-over operation is then performed.

Detailed Description Text (71):
Next, the CVT speed ratio Ic is fixed at Icb', then disengaging of the direct clutch 10 and engaging of the power recirculation clutch 9 are performed. As a result, the IVT speed ratio Ii further increases from Iid' to Icb'. In other words, the IVT speed ratio Ii progressively shifts down from the point A to the point B' via the point D'. Therefore, there is no change in shift direction of the IVT during the operation, as shown by the solid line in FIG. 10. On the other hand, when there is a clutch change-over and the CVT speed ratio Ic is then varied from Ica to Icb', the IVT speed ratio Ii reaches the point B' from the point A via the point C in FIG. 8, and the shift direction does change during the operation as shown by the dotted line in FIG. 10.

Detailed Description Text (73):
Next, the CVT speed ratio is fixed at Icb', then engaging of the direct clutch 10 and disengaging of the power clutch 9 are performed. As a result, the IVT speed ratio Ii further decreases from Iib' to Iid'. In other words, the IVT speed ratio Ii progressively shifts up from the point C to the point D' via the point B'. Therefore in this case also, the shift direction of the IVT does not change during the operation. On the other hand, if there is a clutch change-over and the CVT speed ratio Ic is then varied from Ica to Icb', the IVT speed ratio Ii reaches the point D' from the point C via the point A, so the shift direction does change during operation.

Detailed Description Text (74):
When the IVT speed ratio Ii varies beyond the rotation synchronous point RSP, whether to perform clutch change-over first or variation of the CVT speed ratio Ic first is determined as follows:

Detailed Description Text (76):
When the IVT speed ratio Ii is varied, by changing the order of the clutch change-over and CVT speed ratio control based on a comparison of the real CVT speed ratio RIc and target CVT speed ratio TIc, the variation of the IVT speed ratio Ii due to clutch change-over and the variation of the IVT speed ratio Ii due to the variation of the CVT speed ratio Ic can thus be given the same direction. Therefore, a fast mode change-over due to the partially engaged state of the clutches can be performed without giving an uncomfortable feeling to the driver.

Detailed Description Text (77):
Further, the clutch change-over operation and CVT speed ratio control are not performed simultaneously, the computational load on the control unit 80 is small,

and therefore increase in the manufacturing cost of the speed ratio controller is suppressed.

Detailed Description Text (81):
In a first step S121, it is ~~determined whether or not the~~ clutch ~~change-over~~ flag Fc is 1. When the clutch change-over flag Fc is 1, the subroutine proceeds to a step S122. When the clutch change-over flag Fc is 0, the subroutine proceeds to a step S132.

Detailed Description Text (82):
In the step S122, the same clutch change-over operation is performed as in the step S24 of the first embodiment.

Detailed Description Text (83):
In a next step S123, it is determined whether or not the clutch change-over operation has been completed. According to this embodiment, unlike the step S25 of the first embodiment, determination of completion of clutch change-over is performed by the following method.

Detailed Description Text (84):
The power recirculation clutch 9 and direct clutch 10 are both engaged by supplying oil pressure and disengaged by releasing oil pressure. In order to ensure that the clutch is disengaged when the oil pressure is released, the clutches comprise a return spring which presses a clutch member in the disengaging direction against an oil pressure.

Detailed Description Text (85):
Hence, the completion of clutch change-over is determined by monitoring an oil pressure of a clutch which shifts from the disengaged state to the engaged state. For example, in the case where the IVT shifts up, when an oil pressure Phc of the direct clutch 10 is equal to or greater than a pressure KPrtn of the return spring and a difference in rotation speeds of the two clutches is 0, it is determined that clutch change-over is complete. In the case where the IVT shifts down, when an oil pressure PIc of the power recirculation clutch 9 is equal to or greater than the pressure KPrtn of the turn spring and the difference in rotation speeds of the two clutches is 0, it is determined that clutch change-over is complete. The oil pressures of the clutches are effectively equal to the torques transmitted by the clutches, and when the oil pressure is lower than the return spring pressure KPrtn, the transmitted torque is zero.

Detailed Description Text (86):
Herein, the oil pressure PIc of the power recirculation clutch 9 and the oil pressure Phc of the direct clutch 10 may be found from the duty ratio of the duty signals output to the solenoids 91, 92 from the control unit 80.

Detailed Description Text (87):
The rotation speeds of the clutches 9 and 10 mean the rotation speeds of members which receive the torque via the clutches 9 and 10. In the power recirculation mode, the rotation speed of the power recirculation clutch 9 is represented by the rotation speed of the planet carrier 5B, and the rotation speed of the direct mode clutch 10 is represented by the rotation speed of the CVT output shaft 4.

Detailed Description Text (88):
In the direct mode, the rotation speed of the power recirculation clutch 9 is represented by the rotation speed of the gear output shaft 3C of the reduction gear, and the rotation speed of the direct clutch 10 is represented by the rotation speed of the final output shaft 6. Of these, the rotation speed No of the CVT output shaft 4 and the rotation speed Nout of the final output shaft 6 may be found directly from the output signals of the rotation speed sensor 82 and vehicle speed sensor 83. The rotation speed of the gear output shaft 3C may be found by dividing

a rotation speed Nin of the input shaft 1 detected by the rotation speed sensor 81, by the speed ratio of the reduction gear 3. Further, the rotation speed of the planet carrier 5B may be found by calculation from the rotation speed No of the CVT output shaft 4 and the rotation speed Nout of the final output shaft 6.

Detailed Description Text (89):
As a result of the determination of the step S123, when clutch change-over has not been completed, the subroutine proceeds to a step S126. When clutch change-over has been completed, the subroutine resets the clutch change-over flag Fc to 0 in a step S124, resets the CVT speed ratio control flag FIc to 1 in a step S125, and proceeds to the step S126. When clutch change-over is not complete, the clutch change-over flag Fc remains at 1, so the clutch change-over operation of the step S122 is again performed on the next occasion that the subroutine is executed.

Detailed Description Text (93):
In the step S129, it is determined whether or not the oil pressure of the power recirculation clutch 9 has fallen sufficiently. If the oil pressure of the power recirculation clutch 9 has not fallen sufficiently and CVT speed ratio control is begun, the IVT speed ratio Ii may momentarily vary in an opposite direction to the intended direction due to the transmission torque of the power recirculation clutch 9. To prevent this behavior of the IVT, in the step S129, it is determined whether or not the oil pressure PIc of the power recirculation clutch 9 has become equal to or less than the return spring pressure KPrtn. Also in the step S130, by the same reasoning, it is determined whether or not the oil pressure Phc of the direct clutch 10 has become equal to or less than the return spring pressure KPrtn.

Detailed Description Text (94):
In the step S129 or the step S130, when the clutch pressure exceeds the return spring pressure KPrtn, the subroutine is terminated without performing other steps. As a result, the same determination is again performed on the next occasion that the subroutine is executed, and the CVT speed ratio control of the step S131 and subsequent steps is not performed until the clutch pressure becomes equal to or less than the return spring pressure KPrtn.

Detailed Description Text (95):
In the step S129 or the step S130, when the clutch pressure has become equal to or less than the return spring pressure KPrtn, the subroutine sets the CVT speed ratio control flag FIc to 1 in the step S131, and proceeds to the CVT speed ratio control of the step S132 and subsequent steps.

Detailed Description Text (98):
In this embodiment also, when there is a running mode change-over accompanying shift-up of the CVT 2, a clutch change-over operation is first begun, and if the difference IVTerr between the target IVT speed ratio TIiO and real IVT speed ratio RIi is larger than the predetermined value Kerr, the CVT speed ratio control is performed in parallel with the clutch change-over operation. In other words, by giving priority to the clutch change-over operation, change in the shift direction of the IVT is prevented, and by performing the CVT speed ratio control in parallel with the clutch change-over operation if necessary, delay in the speed ratio variation is prevented and a fast response is obtained.

Detailed Description Text (99):
The predetermined value Kerr which is a reference for determining the speed ratio variation delay represents an allowable maximum discrepancy between the target IVT speed ratio TIiO and real IVT speed ratio RIi to ensure that the vehicle runs without giving an uncomfortable feeling to the driver, and it is found by experiment.

Detailed Description Text (101):
The control unit 80 first starts to engage the power recirculation clutch 9 and

disengage the direct clutch 10 while the CVT speed ratio Ic is fixed at Ica. As a result, the real IVT speed ratio varies from the point A to the point C, but during this process, the difference IVTerr exceeds the predetermined value Kerr. Further, when the clutch pressure of the power recirculation clutch 9 has become equal to or less than the return spring pressure KPrtn at the point .EPSILON., the control unit 80 begins CVT speed ratio control in parallel with the clutch change-over operation by outputting a signal corresponding to the target CVT speed ratio TIc to the step motor 36.

Detailed Description Text (103):
Next, the case will be considered where the IVT shifts up from the point C to the point D in FIG. 8. In this case, the control unit 80 first starts to disengage the power recirculation clutch 9 and engage the direct clutch 10 while the CVT speed ratio Ic is fixed at Ica.

Detailed Description Text (104):
If the IVT speed ratio Ii exceeds the predetermined value Kerr during this operation, CVT speed ratio control is begun by outputting a signal corresponding to the target CVT speed ratio TIc to the step motor 36 when the clutch pressure of the power recirculation clutch 9 has become equal to or less than the return spring pressure KPrtn. As a result, the IVT speed ratio varies from the point .EPSILON. to the point D without passing through the point A. Therefore, the IVT speed ratio Ii continues to follow the target value of the IVT speed ratio during the speed change control without any reversal of direction, and a rapid speed change is achieved without giving an uncomfortable feeling to the driver.

Detailed Description Text (108):
In a next step S143, in the same way as in the step S38, it is determined whether or not the CVT speed ratio control is complete. When the CVT speed ratio control is not complete, the subroutine proceeds to a step S146. When the CVT speed ratio control is complete, the subroutine resets the CVT speed ratio control flag FIc to 0 in a step S144, sets the clutch change-over flag Fc to 1 in a step S145, and proceeds to a step S146. When the CVT speed ratio control is not complete, the CVT speed ratio control flag FIc remains set at 1, so the CVT speed ratio control is again continued in the step S142 on the next occasion that the subroutine is executed.

Detailed Description Text (109):
In the step S146, it is determined whether or not the clutch change-over flag Fc is 1. When the clutch change-over flag FC is 1, the subroutine proceeds to a step S151, and a clutch change-over operation described later is performed.

Detailed Description Text (110):
On the other hand, when the clutch change-over flag Fc is 0, the subroutine proceeds to a step S147.

Detailed Description Text (114):
In the step S150, the clutch change-over flag Fc is set to 1, and the routine proceeds to the clutch change-over operation of the step S151.

Detailed Description Text (117):
In the step S149, when the step motor target operation speed VStp does not exceed the operation speed limit Vmax of the step motor 36, the IVT speed ratio difference IVTerr can be canceled out by controlling only the step motor 36. Therefore, the subroutine is terminated without performing the clutch change-over operation.

Detailed Description Text (118):
In the step S151, the same clutch change-over operation is performed as in the step S34 of the first embodiment.

Detailed Description Text (119):
In a next step S152, whether or not the clutch change-over operation is complete is determined by the same method as that of the step S123. When the clutch change-over operation is not complete, the subroutine is terminated without performing other steps. In this case, the clutch change-over flag Fc remains set at 1, so the clutch change-over operation of the step S151 is again continued on the next occasion that the subroutine is executed. On the other hand, when the clutch change-over operation is complete in the step S152, the routine proceeds to a step S153, the clutch change-over flag Fc is reset to 0, and the subroutine is terminated.

Detailed Description Text (121):
In such a case, if the step motor target operation speed VStp also exceeds the operation speed limit Vmax of the step motor 36, the clutch change-over operation begins in parallel with the CVT speed ratio control.

Detailed Description Text (123):
At the point A in this process, the difference IVTerr of the target IVT speed ratio TIiO and real IVT speed ratio RIi is greater than the predetermined value Kerr. If the step motor target operation speed VStp is larger than the operation speed limit Vmax of the step motor 36 at this time, engaging of the power recirculation clutch 9 and disengaging of the direct clutch 10 begin.

Detailed Description Text (125):
Further, when the IVT speed ratio Ii shifts up from the point B to the point D' in FIG. 8, the control unit 80 first increases the CVT speed ratio Ic from Icb to Icb' by outputting a signal corresponding to the target CVT speed ratio TIc to the step motor 36. At the point C in this process, the difference IVTerr of the target IVT speed ratio TIiO and real IVT speed ratio RIi is greater than the predetermined value Kerr. If the step motor target operation speed VStp is larger than the operation speed limit Vmax of the step motor 36 at this time, disengaging of the power recirculation clutch 9 and engaging of the direct clutch 10 begin.

Detailed Description Text (127):
According to this embodiment, the conditions for starting clutch change-over in parallel with the CVT speed change control are that the difference IVTerr between the target IVT speed ratio TIiO and real IVT speed ratio RIi is larger than the predetermined value Kerr, and that the target operation speed VStp is larger than the operation speed limit Vmax of the step motor 36.

Detailed Description Text (130):
Subsequently, when the target operation speed VStp exceeds the operation speed limit Vmax of the step motor 36 at a time T4, the real IVT speed ratio RIi can no longer be made to follow the target IVT speed ratio TIiO by performing the CVT speed ratio control by outputting a signal to the step motor 36 alone, and the difference IVTerr increases. When the difference IVTerr reaches the predetermined value Kerr at a time T5, engaging of the power recirculation clutch 9 and disengaging of the direct clutch 10 begin. Due to this clutch change-over operation the difference IVTerr is eliminated, and a rapid speed change to the final IVT speed ratio TIc is achieved.

Detailed Description Text (134):
Therefore, when the difference IVTerr of the target IVT speed ratio TIiO and real IVT speed ratio RIi exceeds the predetermined value Kerr and the target operation speed VStp is larger than the operation speed limit Vmax of the step motor 36, a clutch change-over operation is started parallel to the CVT speed ratio control.

Detailed Description Text (135):
For example, if the difference IVTerr exceeds the predetermined value Kerr at the point A of FIG. 8, and the target operation speed VStp at this time exceeds the operation speed limit Vmax of the step motor 36, engaging of the power

recirculation clutch 9 and disengaging of the direct clutch 10 are started at the point A. As a result, the IVT speed ratio Ii varies from the point A to the point B' without passing through the point D'. In this way, the clutch change-over operation compensates the delay of the CVT speed ratio control, and the ability to follow the target IVT speed ratio TIiO improves.

Detailed Description Text (136):
Hence, it is possible to follow the rapid variation of the target IVT speed ratio TIiO, to follow the rapid speed change due to kickdown, and to improve the drivability of the vehicle.

Detailed Description Text (138):
According to this embodiment also, one of the clutch change-over and the control of CVT speed ratio Ic are performed first according to the variation direction of the CVT speed ratio Ic, so the computational load on the control unit 80 is suppressed relatively small.

Detailed Description Paragraph Table (1):
CVT/ IVT shift-up shift-down shift-up clutch change-over CVT speed ratio control shift-down clutch change-over CVT speed ratio control

Current US Cross Reference Classification (2):
701/67

CLAIMS:

1. A speed change controller for an infinite speed ratio continuously variable transmission for use with a vehicle, the infinite speed ratio continuously variable transmission comprising an input shaft, a continuously variable transmission which transmits a rotation speed of the input shaft to a continuously variable transmission output shaft at an arbitrary speed ratio, a fixed speed ratio transmission which transmits the rotation speed of the input shaft to a fixed speed ratio transmission output shaft at a fixed speed ratio, a planetary gear set comprising a first rotation member joined to the continuously variable transmission output shaft, a second rotation member joined to the fixed speed ratio transmission output shaft, and a third rotation member which varies a rotation direction and a rotation speed according to a difference between a rotation speed of the first rotation member and a rotation speed of the second rotation member, a direct clutch which connects and disconnects the continuously variable transmission output shaft and the third rotation member, and a power recirculation clutch which connects and disconnects the fixed speed ratio transmission output shaft and the second rotation member, the controller comprising:

a sensor which detects a running state of the vehicle;

a sensor which detects a real speed ratio of the infinite speed ratio continuously variable transmission; and

a microprocessor programmed to:

set a target speed ratio of the infinite speed ratio continuously variable transmission based on the running state;

determine whether or not the target speed ratio has varied beyond a predetermined speed ratio;

determine which one of control over a combination of the power recirculation clutch and the direct clutch and control over speed ratio will be given priority, depending on both the real speed ratio and the target speed ratio, so that speed ratio control will be implemented continuously in a direction from the real speed

ratio toward the target speed ratio; and

perform control of the power recirculation clutch and direct clutch, and of the speed ratio of the continuously variable transmission, according to the priority.

3. The speed change controller as defined in claim 1, wherein the microprocessor is further programmed to calculate a difference between the target speed ratio and the real speed ratio, and perform the control of the speed ratio of the continuously variable transmission simultaneously with the control of the power recirculation clutch and direct clutch when the difference is larger than a predetermined difference.

4. The speed change controller as defined in claim 3, wherein the microprocessor is further programmed to engage the power recirculation clutch and disengage the direct clutch when the target speed ratio is larger than a predetermined speed ratio, and disengage the power recirculation clutch and engage the direct clutch when the target speed ratio is smaller than the predetermined speed ratio.

5. The speed change controller as defined in claim 4, wherein the microprocessor is further programmed to calculate a connecting pressure of the power recirculation clutch and a connecting pressure of the direct clutch, prohibit control of the speed ratio of the continuously variable transmission until the connecting pressure of the direct clutch becomes equal to or less than a predetermined pressure when the real speed ratio of the infinite speed ratio continuously variable transmission is varying in an increasing direction toward the target speed ratio, and prohibit control of the speed ratio of the continuously variable transmission until the connecting pressure of the power recirculation clutch becomes equal to or less than a predetermined pressure when the real speed ratio of the infinite speed ratio continuously variable transmission is varying in a decreasing direction toward the target speed ratio.

6. The speed change controller as defined in claim 1, wherein the speed change controller further comprises an actuator which varies the speed ratio of the continuously variable transmission according to an output signal from the microprocessor, and the microprocessor is further programmed to calculate a target operation speed of the actuator based on a variation of the target speed ratio, and perform the control of the speed ratio of the continuously variable transmission simultaneously with the control of the power recirculation clutch and direct clutch when the target drive speed is larger than a predetermined limiting speed.

7. The speed change controller as defined in claim 1, wherein the microprocessor is further programmed to give a higher priority to the control of the power recirculation clutch and direct clutch than to the control of the speed ratio of the continuously variable transmission when the speed ratio of the continuously variable transmission is controlled in a decreasing direction, and give a higher priority to the control of the speed ratio of the continuously variable transmission than to the control of the power recirculation clutch and direct clutch when the speed ratio of the continuously variable transmission is controlled in an increasing direction.

8. The speed change controller as defined in claim 1, wherein the running state detection sensor comprises a sensor which detects a running speed of the vehicle, and a sensor which detects a depression amount of an accelerator pedal with which the vehicle is provided.

9. A speed change controller for an infinite speed ratio continuously variable transmission for use with a vehicle, the infinite speed ratio continuously variable transmission comprising an input shaft, a continuously variable transmission which transmits a rotation speed of the input shaft to a continuously variable transmission output shaft at an arbitrary speed ratio, a fixed speed ratio

transmission which transmits the rotation speed of the input shaft to a fixed speed ratio transmission output shaft at a fixed speed ratio, a planetary gear set comprising a first rotation member joined to the continuously variable transmission output shaft, a second rotation member joined to the fixed speed ratio transmission output shaft, and a third rotation member which varies a rotation direction and a rotation speed according to a difference between a rotation speed of the first rotation member and a rotation speed of the second rotation member, a direct clutch which connects and disconnects the continuously variable transmission output shaft and the third rotation member, and a power recirculation clutch which connects and disconnects the fixed speed ratio transmission output shaft and the second rotation member, the controller comprising:

means for detecting a running state of the vehicle;

means for detecting a real speed ratio of the infinite speed ratio continuously variable transmission;

means for setting a target speed ratio of the infinite speed ratio continuously variable transmission based on the running state;

means for determining whether or not the target speed ratio has varied beyond a predetermined speed ratio;

means for determining a priority of controlling the power recirculation clutch and direct clutch, and controlling the speed ratio of the continuously variable transmission, so that, when the target speed ratio has varied beyond the predetermined speed ratio, the real speed ratio of the infinite speed ratio continuously variable transmission varies in the same direction until the real speed ratio reaches the target speed ratio; and

means for performing control of the power recirculation clutch and direct clutch, and of the speed ratio of the continuously variable transmission, according to the priority.

10. A speed change control method for an infinite speed ratio continuously variable transmission for use with a vehicle, the infinite speed ratio continuously variable transmission comprising an input shaft, a continuously variable transmission which transmits a rotation speed of the input shaft to a continuously variable transmission output shaft at an arbitrary speed ratio, a fixed speed ratio transmission which transmits the rotation speed of the input shaft to a fixed speed ratio transmission output shaft at a fixed speed ratio, a planetary gear set comprising a first rotation member joined to the continuously variable transmission output shaft, a second rotation member joined to the fixed speed ratio transmission output shaft, and a third rotation member which varies a rotation direction and a rotation speed according to a difference between a rotation speed of the first rotation member and a rotation speed of the second rotation member, a direct clutch which connects and disconnects the continuously variable transmission output shaft and the third rotation member, and a power recirculation clutch which connects and disconnects the fixed speed ratio transmission output shaft and the second rotation member, the method comprising:

detecting a running state of the vehicle;

detecting a real speed ratio of the infinite speed ratio continuously variable transmission;

setting a target speed ratio of the infinite speed ratio continuously variable transmission based on the running state;

determining whether or not the target speed ratio has varied beyond a predetermined

speed ratio;

determining a priority of controlling the power recirculation clutch and direct clutch, and controlling the speed ratio of the continuously variable transmission, so that, when the target speed ratio has varied beyond the predetermined speed ratio, the real speed ratio of the infinite speed ratio continuously variable transmission varies in the same direction until the real speed ratio reaches the target speed ratio; and

performing control of the power recirculation clutch and direct clutch, and of the speed ratio of the continuously variable transmission, according to the priority.

11. A speed change controller for an infinite speed ratio continuously variable transmission for use with a vehicle, the infinite speed ratio continuously variable transmission comprising an input shaft, a continuously variable transmission which transmits a rotation speed of the input shaft to a continuously variable transmission output shaft at an arbitrary speed ratio, a fixed speed ratio transmission which transmits the rotation speed of the input shaft to a fixed speed ratio transmission output shaft at a fixed speed ratio, a planetary gear set comprising a first rotation member joined to the continuously variable transmission output shaft, a second rotation member joined to the fixed speed ratio transmission output shaft, and a third rotation member which varies a rotation direction and a rotation speed according to a difference between a rotation speed of the first rotation member and a rotation speed of the second rotation member, a direct clutch which connects and disconnects the continuously variable transmission output shaft and the third rotation member, and a power recirculation clutch which connects and disconnects the fixed speed ratio transmission output shaft and the second rotation member, the controller comprising:

a sensor which detects a running state of the vehicle;

a sensor which detects a real speed ratio of the infinite speed ratio continuously variable transmission; and

a microprocessor programmed to:

set a target speed ratio of the infinite speed ratio continuously variable transmission based on the running state;

determine if a mode change-over between a power recirculation mode wherein the power recirculation clutch is engaged while the direct clutch is disengaged, and a direct mode wherein the direct clutch is engaged while the power recirculation clutch is disengaged, is required to achieve the target speed ratio;

determine which one of control over a combination of the power recirculation clutch and the direct clutch and control over speed ratio will be given priority, depending on both the real speed ratio and the target speed ratio, so that speed ratio control will be implemented continuously in a direction from the real speed ratio toward the target speed ratio; and

perform control of the power recirculation clutch and direct clutch, and of the speed ratio of the continuously variable transmission, according to the determined priority.

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L9: Entry 1 of 1 File: USPT May 24, 2005

US-PAT-NO: 6898504
DOCUMENT-IDENTIFIER: US 6898504 B2

TITLE: Vehicle driving force control apparatus

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PRIOR-ART-DISCLOSED:

U.S. PATENT DOCUMENTS

Search Selected Search ALL Clear

	PAT-NO	ISSUE-DATE	PATENTEE-NAME	US-CL
☐	4180138	December 1979	Shea	
☐	5036718	August 1991	Bulgrien	
☐	6321865	November 2001	Kuribayashi et al.	180/243
☐	6434469	August 2002	Shimizu et al.	
☐	6442454	August 2002	Akiba et al.	

☐	6464608	October 2002	Bowen et al.	475/5
☐	6606549	August 2003	Murakami et al.	701/89
☐	2002/0107101	August 2002	Bowen et al.	
☐	2003/0010559	January 2003	Suzuki	
☐	2003/0064858	April 2003	Saeki et al.	
☐	2003/0089539	May 2003	Kadota	
☐	2003/0151381	August 2003	Kadota et al.	

FOREIGN PATENT DOCUMENTS

FOREIGN-PAT-NO	PUBN-DATE	COUNTRY	US-CL
0 241 215	October 1987	EP	
0 314 452	May 1989	EP	
0 799 740	October 1997	EP	
0 963 892	December 1999	EP	
1 127 735	August 2001	EP	
1 226 993	July 2002	EP	
11-243608	September 1999	JP	
2001-138764	May 2001	JP	
2001-146930	May 2001	JP	
2002-200932	July 2002	JP	
2002-218605	August 2002	JP	
2003-025861	January 2003	JP	
2003-130200	May 2003	JP	
2003-156079	May 2003	JP	
2003-209902	July 2003	JP	
WO 02-064996	August 2002	WO	
WO 02-087916	November 2002	WO	

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ATTY-AGENT-FIRM: Shinjyu Global IP Counselors, LLP

ABSTRACT:

A vehicle driving force control apparatus is configured for a four-wheel drive vehicle that switches between a four-wheel drive state and a two-wheel drive state. The driving force control apparatus improves the response when a vehicle shifts into a four-wheel drive state while starting to move from a state of rest and, simultaneously, avoids the occurrence of shock when the clutch is connected. When the rotational speeds of the motor and the rear wheels reach or fall below their respective minimum detectable rotational speeds, the controller calculates estimated times until the motor and the rear wheels will stop based on the rotational speeds detected up until the minimum detectable rotational speed was reached and begins counting down from those estimated times. Since the clutch is connected after both estimated times have reached zero, the clutch is connected

while reliably avoiding the occurrence of shock.

21 Claims, 18 Drawing figures

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L18: Entry 1 of 1 File: USPT Feb 26, 2002

DOCUMENT-IDENTIFIER: US 6351700 B1
TITLE: Speed change controller and control method of infinite speed ratio continuously variable transmission

Brief Summary Text (13):
In order to achieve the above object, this invention provides a speed change controller for such an infinite speed ratio continuously variable transmission for use with a vehicle that comprises an input shaft, a continuously variable transmission which transmits a rotation speed of the input shaft to a continuously variable transmission output shaft at an arbitrary speed ratio, a fixed speed ratio transmission which transmits the rotation speed of the input shaft to a fixed speed ratio transmission output shaft at a fixed speed ratio, a planetary gear set comprising a first rotation member joined to the continuously variable transmission output shaft, a second rotation member joined to the fixed speed ratio transmission output shaft, and a third rotation member which varies a rotation direction and a rotation speed according to a difference between a rotation speed of the first rotation member and a rotation speed of the second rotation member, a direct clutch which connects and disconnects the continuously variable transmission output shaft and the third rotation member, and a power recirculation clutch which connects and disconnects the fixed speed ratio transmission output shaft and the second rotation member.

CLAIMS:

1. A speed change controller for an infinite speed ratio continuously variable transmission for use with a vehicle, the infinite speed ratio continuously variable transmission comprising an input shaft, a continuously variable transmission which transmits a rotation speed of the input shaft to a continuously variable transmission output shaft at an arbitrary speed ratio, a fixed speed ratio transmission which transmits the rotation speed of the input shaft to a fixed speed ratio transmission output shaft at a fixed speed ratio, a planetary gear set comprising a first rotation member joined to the continuously variable transmission output shaft, a second rotation member joined to the fixed speed ratio transmission output shaft, and a third rotation member which varies a rotation direction and a rotation speed according to a difference between a rotation speed of the first rotation member and a rotation speed of the second rotation member, a direct clutch which connects and disconnects the continuously variable transmission output shaft and the third rotation member, and a power recirculation clutch which connects and disconnects the fixed speed ratio transmission output shaft and the second rotation member, the controller comprising:

a sensor which detects a running state of the vehicle;

a sensor which detects a real speed ratio of the infinite speed ratio continuously variable transmission; and

a microprocessor programmed to:

set a target speed ratio of the infinite speed ratio continuously variable transmission based on the running state;

determine whether or not the target speed ratio has varied beyond a predetermined speed ratio;

determine which one of control over a combination of the power recirculation clutch and the direct clutch and control over speed ratio will be given priority, depending on both the real speed ratio and the target speed ratio, so that speed ratio control will be implemented continuously in a direction from the real speed ratio toward the target speed ratio; and

perform control of the power recirculation clutch and direct clutch, and of the speed ratio of the continuously variable transmission, according to the priority.

5. The speed change controller as defined in claim 4, wherein the microprocessor is further programmed to calculate a connecting pressure of the power recirculation clutch and a connecting pressure of the direct clutch, prohibit control of the speed ratio of the continuously variable transmission until the connecting pressure of the direct clutch becomes equal to or less than a predetermined pressure when the real speed ratio of the infinite speed ratio continuously variable transmission is varying in an increasing direction toward the target speed ratio, and prohibit control of the speed ratio of the continuously variable transmission until the connecting pressure of the power recirculation clutch becomes equal to or less than a predetermined pressure when the real speed ratio of the infinite speed ratio continuously variable transmission is varying in a decreasing direction toward the target speed ratio.

9. A speed change controller for an infinite speed ratio continuously variable transmission for use with a vehicle, the infinite speed ratio continuously variable transmission comprising an input shaft, a continuously variable transmission which transmits a rotation speed of the input shaft to a continuously variable transmission output shaft at an arbitrary speed ratio, a fixed speed ratio transmission which transmits the rotation speed of the input shaft to a fixed speed ratio transmission output shaft at a fixed speed ratio, a planetary gear set comprising a first rotation member joined to the continuously variable transmission output shaft, a second rotation member joined to the fixed speed ratio transmission output shaft, and a third rotation member which varies a rotation direction and a rotation speed according to a difference between a rotation speed of the first rotation member and a rotation speed of the second rotation member, a direct clutch which connects and disconnects the continuously variable transmission output shaft and the third rotation member, and a power recirculation clutch which connects and disconnects the fixed speed ratio transmission output shaft and the second rotation member, the controller comprising:

means for detecting a running state of the vehicle;

means for detecting a real speed ratio of the infinite speed ratio continuously variable transmission;

means for setting a target speed ratio of the infinite speed ratio continuously variable transmission based on the running state;

means for determining whether or not the target speed ratio has varied beyond a predetermined speed ratio;

means for determining a priority of controlling the power recirculation clutch and direct clutch, and controlling the speed ratio of the continuously variable transmission, so that, when the target speed ratio has varied beyond the predetermined speed ratio, the real speed ratio of the infinite speed ratio

continuously variable transmission varies in the same direction until the real speed ratio reaches the target speed ratio; and

means for performing control of the power recirculation clutch and direct clutch, and of the speed ratio of the continuously variable transmission, according to the priority.

10. A speed change control method for an infinite speed ratio continuously variable transmission for use with a vehicle, the infinite speed ratio continuously variable transmission comprising an input shaft, a continuously variable transmission which transmits a rotation speed of the input shaft to a continuously variable transmission output shaft at an arbitrary speed ratio, a fixed speed ratio transmission which transmits the rotation speed of the input shaft to a fixed speed ratio transmission output shaft at a fixed speed ratio, a planetary gear set comprising a first rotation member joined to the continuously variable transmission output shaft, a second rotation member joined to the fixed speed ratio transmission output shaft, and a third rotation member which varies a rotation direction and a rotation speed according to a difference between a rotation speed of the first rotation member and a rotation speed of the second rotation member, a direct clutch which connects and disconnects the continuously variable transmission output shaft and the third rotation member, and a power recirculation clutch which connects and disconnects the fixed speed ratio transmission output shaft and the second rotation member, the method comprising:

detecting a running state of the vehicle;

detecting a real speed ratio of the infinite speed ratio continuously variable transmission;

setting a target speed ratio of the infinite speed ratio continuously variable transmission based on the running state;

determining whether or not the target speed ratio has varied beyond a predetermined speed ratio;

determining a priority of controlling the power recirculation clutch and direct clutch, and controlling the speed ratio of the continuously variable transmission, so that, when the target speed ratio has varied beyond the predetermined speed ratio, the real speed ratio of the infinite speed ratio continuously variable transmission varies in the same direction until the real speed ratio reaches the target speed ratio; and

performing control of the power recirculation clutch and direct clutch, and of the speed ratio of the continuously variable transmission, according to the priority.

11. A speed change controller for an infinite speed ratio continuously variable transmission for use with a vehicle, the infinite speed ratio continuously variable transmission comprising an input shaft, a continuously variable transmission which transmits a rotation speed of the input shaft to a continuously variable transmission output shaft at an arbitrary speed ratio, a fixed speed ratio transmission which transmits the rotation speed of the input shaft to a fixed speed ratio transmission output shaft at a fixed speed ratio, a planetary gear set comprising a first rotation member joined to the continuously variable transmission output shaft, a second rotation member joined to the fixed speed ratio transmission output shaft, and a third rotation member which varies a rotation direction and a rotation speed according to a difference between a rotation speed of the first rotation member and a rotation speed of the second rotation member, a direct clutch which connects and disconnects the continuously variable transmission output shaft and the third rotation member, and a power recirculation clutch which connects and disconnects the fixed speed ratio transmission output shaft and the second rotation

member, the controller comprising:

a sensor which detects a running state of the vehicle;

a sensor which detects a real speed ratio of the infinite speed ratio continuously variable transmission; and

a microprocessor programmed to:

set a target speed ratio of the infinite speed ratio continuously variable transmission based on the running state;

determine if a mode change-over between a power recirculation mode wherein the power recirculation clutch is engaged while the direct clutch is disengaged, and a direct mode wherein the direct clutch is engaged while the power recirculation clutch is disengaged, is required to achieve the target speed ratio;

determine which one of control over a combination of the power recirculation clutch and the direct clutch and control over speed ratio will be given priority, depending on both the real speed ratio and the target speed ratio, so that speed ratio control will be implemented continuously in a direction from the real speed ratio toward the target speed ratio; and

perform control of the power recirculation clutch and direct clutch, and of the speed ratio of the continuously variable transmission, according to the determined priority.

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DOCUMENT-IDENTIFIER: US 6898504 B2
TITLE: Vehicle driving force control apparatus

Abstract Text (1):
A vehicle driving force control apparatus is configured for a four-wheel drive vehicle that switches between a four-wheel drive state and a two-wheel drive state. The driving force control apparatus improves the response when a vehicle shifts into a four-wheel drive state while starting to move from a state of rest and, simultaneously, avoids the occurrence of shock when the clutch is connected. When the rotational speeds of the motor and the rear wheels reach or fall below their respective minimum detectable rotational speeds, the controller calculates estimated times until the motor and the rear wheels will stop based on the rotational speeds detected up until the minimum detectable rotational speed was reached and begins counting down from those estimated times. Since the clutch is connected after both estimated times have reached zero, the clutch is connected while reliably avoiding the occurrence of shock.

Brief Summary Text (3):
The present invention generally relates to a vehicle driving force control apparatus for a four-wheel drive vehicle. More specifically, the present invention relates to a vehicle driving force control apparatus for a four-wheel drive vehicle that switches between a four-wheel drive state and a two-wheel drive state in response to such factors as a traveling state.

Brief Summary Text (5):
An example of this kind of four-wheel drive vehicle driving force control apparatus is disclosed in Japanese Laid-Open Patent Publication No. 11-243608. In the vehicle described in that publication, the front wheels are the main drive wheels that are driven by an internal combustion engine, while the rear wheels are the subordinate drive wheels that are driven by an electric motor. When the vehicle is in a four-wheel drive state, both the front and rear wheels are driven together. A clutch and a reduction gear are installed in the torque transfer path between the electric motor and the rear wheel axle. The technology disclosed in this publication employs a drive control method in which the electric motor is rotated in an unloaded state until it reaches a rotational speed equivalent to the rotational speed of the rear wheel axle before connecting the clutch. After the clutch is connected, the output torque of the electric motor is then increased gradually.

Brief Summary Text (6):
In view of the above, it will be apparent to those skilled in the art from this disclosure that there exists a need for an improved vehicle driving force control apparatus. This invention addresses this need in the art as well as other needs, which will become apparent to those skilled in the art from this disclosure.

Brief Summary Text (8):
It has been discovered that the drive control method just described results in a the delay in the response of the vehicle to a driver's request to start moving or to accelerate in situations where acceleration slippage occurs easily, such as when the vehicle is initially starting to move from a state of rest. In particular, result is due to the fact that the connection of the clutch and the gradual

increasing of the motor torque are conducted after acceleration slippage of the front wheels (main drive wheels) is detected.

Brief Summary Text (9):
One feasible method to improve the vehicle response when the vehicle starts to move from a rest state is to connect the clutch while the vehicle is stopped (i.e., before the vehicle starts to move). This enables the required motor torque to be applied from the initial stage of starting to move while avoiding the occurrence of shock when the clutch is connected.

Brief Summary Text (10):
A feasible way to determine if the vehicle is stopped is to detect the wheel speed, but it is possible that the electric motor and the input side of clutch will still be rotating when the wheels stop (vehicle stops) immediately after shifting from a four-wheel drive state to a two-wheel drive state. Consequently, even though the vehicle is determined to be in a stopped state based on the wheel speed, there is the risk that a rotational speed difference greater than or equal to a prescribed value will still exist between the input shaft and the output shaft of the clutch, thus causing shock to occur when the clutch is connected.

Brief Summary Text (11):
Moreover, rotation sensors that detect the rotational speed of wheels and the like generally are not able to detect accurately when the rotational speed is very low because the magnetic flux pulse is small. In other words, in the course of decelerating to a stopped state, the wheels of a vehicle enter a region of very slow rotation where the rotation sensor is substantially unable to detect the rotational speed. Thus, in this situation, it is impossible to determine when the vehicle has stopped. Consequently, there is a problem in that it cannot be determined if the clutch should be connected.

Brief Summary Text (12):
The present invention was conceived in view of these problems and its object is to provide a four-wheel drive vehicle driving force control apparatus that can improve the vehicle response with which the vehicle shifts to a four-wheel drive state when the vehicle is starting to move while avoiding the occurrence of shock when the clutch is connected.

Brief Summary Text (13):
In order to achieve the aforementioned object, the present invention provides a vehicle driving force control apparatus for a vehicle power train having a clutch installed in a torque transfer path from a drive source to a wheel, the clutch having an input part connected to the drive source and an output part connected to the wheel. The vehicle driving force control apparatus basically comprises an output rotational speed sensor, an input rotational speed sensor, an output stop estimating section, an input stop estimating section, a vehicle stop determining section, a clutch stop determining section and a clutch connection command outputting section. The output rotational speed sensor is configured to detect an output rotational speed of the output part of the clutch and produce an output rotational speed value. The input rotational speed sensor is configured to detect an input rotational speed of the input part of the clutch and produce an input rotational speed value. The output stop estimating section is configured to estimate that the rotation of the output part has stop rotating upon an occurrence of a detected first parameter that is based on the output rotational speed value received from the output rotational speed sensor. The input stop estimating section is configured to estimate that the rotation of the input part has stop rotating upon an occurrence of a detected second parameter that is based on the input rotational speed value received from the input rotational speed sensor. The vehicle stop determining section is configured to a determination whether the vehicle has stopped. The clutch stop determining section is configured to determine that the clutch has stopped rotating based on a determination of the occurrences of the

detected output and input parameters, upon the vehicle stop determining section determining that the vehicle has stopped. The clutch connection command outputting section is configured to output a clutch connection command to connect the clutch, upon the clutch stop determining section determining that the clutch has stopped rotating.

Drawing Description Text (3):
FIG. 1 is a schematic block diagram of a vehicle equipped with a vehicle driving force control apparatus in accordance with one embodiment of the present invention;

Drawing Description Text (4):
FIG. 2 is a block diagram showing a control system configuration for the vehicle driving force control apparatus illustrated in FIG. 1 in accordance with the illustrated embodiment of the present invention;

Drawing Description Text (5):
FIG. 3 is a block diagram showing the 4WD controller for the vehicle driving force control apparatus of the illustrated embodiment of the present invention;

Drawing Description Text (6):
FIG. 4 is a flow chart showing the processing sequence executed by the 4WD controller for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention;

Drawing Description Text (7):
FIG. 5 is a flow chart showing the processing sequence executed by the power supply managing section for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention;

Drawing Description Text (8):
FIG. 6 is a flow chart showing the processing sequence executed by the surplus torque computing section for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention;

Drawing Description Text (9):
FIG. 7 is a flow chart showing the processing sequence executed by the target torque control section for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention;

Drawing Description Text (10):
FIG. 8 is a flow chart showing the processing sequence executed by the surplus torque converting section for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention;

Drawing Description Text (11):
FIG. 9 is a flow chart showing the processing sequence executed by the backlash eliminating section for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention;

Drawing Description Text (12):
FIG. 10 is a flow chart showing the processing sequence executed by the clutch connection determining section for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention;

Drawing Description Text (13):
FIG. 11 is a flow chart showing the processing sequence executed by the wheel stop estimating section for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention;

Drawing Description Text (14):
FIG. 12 is a flow chart showing the processing sequence executed by the motor stop estimating section for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention;

Drawing Description Text (15):
FIG. 13 is a flow chart showing the processing sequence executed by the external disturbance detecting section for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention;

Drawing Description Text (16):
FIG. 14 is a flow chart showing the processing sequence executed by the clutch connectability determining section for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention;

Drawing Description Text (17):
FIG. 15 is a flow chart showing the processing sequence executed by the clutch control flag outputting section for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention;

Drawing Description Text (18):
FIG. 16 is a flow chart showing the processing sequence executed by the engine controller for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention;

Drawing Description Text (19):
FIG. 17 are timing charts that illustrate the rotation stop timing estimation executed the rotation stop timing estimation section for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention; and

Drawing Description Text (20):
FIG. 18 are time charts for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention.

Detailed Description Text (3):
Referring initially to FIG. 1, a four wheel drive vehicle is diagrammatically illustrated that is equipped with a vehicle driving force control apparatus in accordance with a first embodiment of the present invention. As shown in FIG. 1, the vehicle in accordance with this embodiment has left and right front wheels 1L and 1R that are driven by an internal combustion engine or main drive source 2, and left and right rear wheels 3L and 3R that are driven by an electric motor or subordinate drive source 4. Thus, the front wheels 1L and 1R serve as the main drive wheels, while the rear wheels 3L and 3R serve as the subordinate drive wheels. An endless drive belt 6 transfers power from the internal combustion engine 2 to a generator 7, which supplies electrical energy to the electric motor 4.

Detailed Description Text (4):
The generator 7 rotates at a rotational speed Nh that is equal to the product of the rotational speed Ne of the internal combustion engine 2 and the pulley ratio of the endless drive belt 6. The load placed on the internal combustion engine 2 by the generator 7 due to the field current Ifh is adjusted by the 4WD controller 8 to generate a voltage corresponding to the load torque. The voltage generated by the generator 7 can be supplied to the electric motor 4 through the electrical line 9. A junction box 10 is provided at an intermediate point in the electrical line 9 between the electric motor 4 and the generator 7. The drive shaft of the electric motor 4 can be connected to the rear wheels 3L and 3R via a reduction gear 11, a clutch 12 and a differential gear 13.

Detailed Description Text (5):

Basically, in the vehicle driving force control apparatus of the present invention, as explained below, when it is estimated that the vehicle will stop from a traveling state, a transition time is estimated based on an estimate of the amount of time until the input shaft of the clutch 12 will stop and an estimate of the amount of time until the output shaft of the clutch 12 will stop, with both estimates being calculated while the vehicle is traveling. The clutch 12 is then connected after the transition time (which generally corresponds to the larger of the two aforementioned estimates) has elapsed. It is assumed that both the input shaft and the output shaft of the clutch 12 have completely stopped after the transition time has elapsed. As a result, it is possible to reliably connect the clutch 12 before the vehicle starts moving again (which is a time when four-wheel drive is often demanded) while avoiding the occurrence of shock when the clutch 12 connects.

Detailed Description Text (6):
These estimated amounts of time are assumed to be estimated based on values detected of the rear wheels 3L and 3R before it was estimated that the traveling vehicle would stop, i.e., before the vehicle reached a very low traveling speed. In short, the estimated amounts of time are assumed to be estimated based on detection values that are greater than or equal to a minimum detectable rotational speed. Consequently, if the fluctuations (external disturbances) of the traveling state are small from the time when it is estimated that the vehicle will stop until the time when the vehicle stops, it is possible to calculate an estimate of the amount of time until each shaft will stop with the required degree of accuracy. These estimated amounts of time are preferably estimated based on the rate of change (particularly the deceleration rate) in the rotation time.

Detailed Description Text (7):
A main throttle valve 15 and a sub throttle valve 16 are disposed inside the intake passage 14 (e.g., an intake manifold) of the internal combustion engine 2. The throttle opening of the main throttle valve 15 is adjusted/controlled in accordance with the amount of depression of the accelerator pedal 17, which also constitutes or functions as an accelerator position or sensor, or a throttle opening instructing device or sensor. In order to adjust the throttle opening of the main throttle valve 15, the main throttle valve 15 is either mechanically linked to the depression amount of the accelerator pedal 17, or adjusted/controlled electrically by an engine controller 18 in accordance with the depression amount detection value from an accelerator sensor 40 that detects the depression amount of the accelerator pedal 17 or the degree of opening of the main throttle valve 15. The depression amount detection value of the accelerator sensor 40 is outputted as a control signal to the 4WD controller 8. The accelerator sensor 40 constitutes an acceleration instruction sensor. The phrase "accelerator position opening degree" as used herein refers to either a throttle opening amount of the main throttle valve 15 or a depression amount of the accelerator pedal 17 or similar accelerator device. Thus, the phrase "accelerator position opening degree" as used herein refers to either a throttle opening amount of the main throttle valve 15 or a depression amount of the accelerator pedal 17 or similar accelerator device.

Detailed Description Text (8):
The sub throttle valve 16 uses a stepper motor 19 as an actuator for adjusting its throttle opening. Specifically, the throttle opening of the sub throttle valve 16 is adjusted/controlled by the rotational angle of the stepper motor 19, which corresponds to the step count. The rotational angle of the stepper motor 19 is adjusted/controlled by a drive signal from the motor controller 20. The sub throttle valve 16 is provided with a throttle sensor. The step count of the stepper motor 19 is feedback-controlled based on the throttle opening detection value detected by this throttle sensor. The output torque of the internal combustion engine 2 can be controlled (reduced) independently of the driver's operation of the accelerator pedal by adjusting the throttle opening of the sub throttle valve 16 so as to be smaller than the throttle opening of the main throttle valve 15.

Detailed Description Text (13):
The brake controller 36 controls the braking force acting on the vehicle by controlling the braking devices (e.g., disc brakes) 37FL, 37FR, 37RL and 37RR installed on the wheels 1L, 1R, 3L and 3R in response to the inputted brake stroke amount.

Detailed Description Text (17):
The vehicle driving force control apparatus is also equipped with a motor rotational speed sensor 26 that detects the rotational speed Nm of the drive shaft of the electric motor 4. The motor rotational speed sensor 26 outputs a control signal indicative of the detected rotational speed of the electric motor 4 to the 4WD controller 8. The motor rotational speed sensor 26 constitutes an input shaft rotational speed detector or sensor.

Detailed Description Text (18):
The clutch 12 is a hydraulic clutch or electric clutch that connects and disconnects in response to a clutch control command issued from the 4WD controller 8. Thus, the clutch 12 transmits torque from the electric motor 4 to the rear wheels 3L and 3R at a torque transfer rate corresponding to the clutch control command from the 4WD controller 8.

Detailed Description Text (20):
The vehicle driving force control apparatus is also equipped with a drive mode switch 42 that allows the driver to manually select either a two-wheel (non-all wheel) drive mode or a four-wheel (all wheel) drive mode. The drive mode switch 42 is configured and arranged to output to the 4WD controller 8 a control signal that is indicative of the selected or designated drive mode to the 4WD controller 8. Thus, the 4WD controller 8 has a clutch connection command outputting section that is configured to output a clutch connection command 12 to connect the clutch 12 when the four-wheel drive mode has been designated. Thus, the drive mode switch 42 of the present invention constitutes part of a drive mode selection section that is configured to select one of a multi-wheel drive mode and a non-all wheel drive mode. When the present invention is utilized in vehicles equipped with more than four wheels or without an all wheel drive mode, the multi-wheel drive mode refers to a mode in which at least one of (main) drive wheel driven by a first (main) drive source is driven and at least one second (subordinate) drive wheel driven by a second (subordinate) drive source with a clutch disposed between the second drive wheel and the second drive source is driven. In this situation, a non-all wheel drive mode refers to a mode in which at least the clutch disconnects the second (subordinate) drive source from the second (subordinate) wheel.

Detailed Description Text (21):
A warning lamp 41 for clutch connection is arranged inside the passenger compartment. The warning lamp 41 either flashes or goes out (does not light) based on a signal from the 4WD controller 8 that indicates that a problem exists or not with the clutch connection.

Detailed Description Text (22):
A 12-volt battery 43 supplies operating electric power to the 4WD controller 8 with a 12-volt relay 44 is installed in the 12-volt electric power supply line thereof in order to connect and disconnect the power to the clutch 12, which is preferably an electromagnetic clutch.

Detailed Description Text (23):
As shown in FIG. 3, the 4WD controller 8 is equipped with a generator control section 8A, a relay control section 8B, a motor control section 8C, a clutch control section 8D, a surplus torque computing section 8E, a target torque limiting section 8F, a surplus torque converting section 8G, a backlash or play elimination control section 8H, an electric power supply managing section 8J, a 12-volt relay

control section 8K, and a clutch connection determining section 8L. The clutch connection determining section 8L constitutes or includes an output shaft stop estimating section or device, an input shaft stop estimating section or device, and a clutch connection command outputting section or device.

Detailed Description Text (24):
The 12-volt relay control section 8K controls the 12-volt relay 44 in order to connect and disconnect the electrical power supplied from the 12-volt battery 43 to the 4WD controller 8, which distributes the electrical power to various components of the vehicle driving force control apparatus. The electric power supply managing section 8J, the surplus torque computing section 8E only operate when electrical power is supplied from the 12-volt battery 43. Of course, the other sections of the 4WD controller 8, with the exception of the 12-volt relay control section 8K, continue to operate even when the electrical power is disconnect from the 12-volt battery 43 to the 4WD controller 8.

Detailed Description Text (25):
Through the voltage adjuster 22, the generator control section 8A monitors the generated voltage V of the generator 7 and adjusts the generated voltage V of the generator 7 to the required voltage by adjusting the field current Ifh of the generator 7. Thus, the generator control section 8A includes a generation load torque adjusting section as discussed below. The relay control section 8B controls shutting off and connecting the power supply from the generator 7 to the electric motor 4. The motor control section 8C adjusts the field current Ifm of the electric motor 4 in order to adjust the torque of the electric motor 4 to the required value. The clutch control section 8D controls the state of the clutch 12 by outputting a clutch control command to the clutch 12.

Detailed Description Text (27):
First, the processing shown in FIG. 5 is executed by the electric power supply managing section 8J. In step S1, mode information is received from the drive mode switch 42, while in step S3, the 4WD controller 8 determines if the vehicle is in four-wheel drive mode or in two-wheel drive mode. If the vehicle is in four-wheel drive mode, the 4WD controller 8 proceeds to step S5. If the vehicle is in two-wheel drive mode, the 4WD controller 8 proceeds to step S7.

Detailed Description Text (28):
In step S5, the 12-volt relay control section 8K outputs the 12-volt relay ON command such that electric power is supplied to activate the clutch 12 and the 4WD controller 8 returns to the beginning of the control loop. Meanwhile, in step S7, the 12-volt relay control section 8K outputs the 12-volt relay OFF command such that electric power is shut off and the 4WD controller 8 returns to the beginning of the control loop. When the electric power is shut off, the surplus torque computing section 8E does not execute any processing thereafter and output of command values to the generator 7 and motor 4 stops.

Detailed Description Text (29):
Next, the surplus torque computing section 8E will be discussed which executes the processing shown in FIG. 6. First, in step S10, the wheel speeds computed based on the signals from the wheel speed sensors 27FL, 27FR, 27RL and 27RR are used to subtract the wheel speed of the rear wheels 3L and 3R (subordinate drive wheels) from the wheel speed of the front wheels 1L and 1R (main drive wheels) and find the slippage speed .DELTA.V.sub.F, which is the magnitude of the acceleration slippage of the front wheels 1L and 1R. Then, the 4WD controller 8 proceeds to step S20.

Detailed Description Text (31):
Now, the slippage speed (acceleration slippage magnitude) .DELTA.V.sub.F of the front or main drive wheels 1L and 1R is calculated by the differential between the average front wheel speed V.sub.Wf and the average rear wheel speed V.sub.Wr, as set forth in the following equation:

Detailed Description Text (32):
In step S20, the 4WD controller 8 determines whether or not the calculated slippage speed .DELTA.V.sub.F exceeds a prescribed value, such as zero. Thus, steps S10 and S20 constitute an acceleration slippage detection section that estimates if acceleration slippage is occurring in the front wheels 1L and 1R that is driven by the internal combustion engine 2. If slippage speed .DELTA.V.sub.F is determined to be zero or below, it is estimated that the front wheels 1L and 1R are not experiencing acceleration slippage and the 4WD controller 8 proceeds to step S30, where a target generator load torque Th is set to zero and the 4WD controller 8 returns to the beginning of the control loop.

Detailed Description Text (33):
Conversely, if in step S20 slippage speed .DELTA.V.sub.F is determined to be larger than zero, it is estimated that the front wheels 1L and 1R are experiencing acceleration slippage, and thus, control proceeds to step S40.

Detailed Description Text (34):
In step S40, the absorption torque T.DELTA.V.sub.F required for suppressing the acceleration slippage of the front wheels 1L and 1R is calculated using the equation below and the 4WD controller 8 proceeds to step S50. The absorption torque T.DELTA.V.sub.F is an amount that is proportional to the acceleration slippage magnitude, as set forth in the following equation:

Detailed Description Text (45):
Next, the processing executed by the target torque limiting section 8F will be explained based on FIG. 7. The processing of the target generator load torque Th in the flow chart of FIG. 7 constitutes a generator control section configured to control a generation load torque of the generator 7 to substantially correspond to an acceleration slippage magnitude of the drive wheel, when the acceleration slippage detection section estimates acceleration slippage occurring in the drive wheel.

Detailed Description Text (51):
Next, the processing executed by the surplus torque converting section 8G of the 4WD controller 8 will be explained based on FIG. 8. First, in step S200, the 4WD controller 8 determines if the target generator load torque Th is larger than 0. If the target generator load torque Th is determined to be larger than 0, then the 4WD controller 8 proceeds to step S210 because the front wheels 1L and 1R are experiencing acceleration slippage. If the 4WD controller 8 determines that the target generator load torque Th is less than or equal to 0, then the 4WD controller 8 proceeds to step S290 because the front wheels 1L and 1R are not experiencing acceleration slippage.

Detailed Description Text (54):
It is also acceptable to provide a motor torque correcting section that continuously corrects the required motor torque Tm by adjusting the field current Ifm in accordance with the rotational speed Nm of the electric motor 4. That is, instead of switching between two stages, the field current Ifm of the electric motor 4 can be adjusted in accordance with the motor rotational speed Nm. As a result, even if the electric motor 4 rotates at a high speed, the required motor torque Tm can be obtained because the motor induced voltage E of the electric motor 4 is kept from rising and the motor torque is prevented from decreasing. Furthermore, since a smooth motor torque characteristic can be obtained, the vehicle can travel with better stability than in the case of two-stage control and the vehicle can always be kept in a state where the motor driving efficiency is good.

Detailed Description Text (70):
In step S460, the corresponding target armature current Gala to be used for

eliminating backlash is calculated using the backlash elimination-purpose target motor torque GaTm as a variable. Then, in step S470, the induced voltage GaE of the electric motor 4 is calculated based on the rotational speed Nm of the electric motor 4 and the motor field current Imf, which has been fixed at a prescribed value. Then, the 4WD controller 8 proceeds to step S480. If the backlash elimination control is only executed when the vehicle is starting to move from a stop, it is acceptable to ignore the fluctuations in the induced voltage GaE of the electric motor 4 and execute the processing using a fixed value for (i.e., without calculating) the induced voltage GaE.

Detailed Description Text (72):
In step S510, the backlash elimination flag GATAFLG is set to 1 and the 4WD controller 8 proceeds to step S520. Setting the backlash elimination flag GATAFLG to 1 causes the surplus torque converting section 8G to process the motor torque in accordance with the outputted target voltage GaV and the target generator load torque GaTh. In short, since backlash elimination processing is in progress, the electric motor 4 is drive-controlled and generates a very small torque after the clutch 12 is connected.

Detailed Description Text (73):
In step 520, the rotational speeds of the axles of the rear wheels (which are the subordinate drive wheels) 3L and 3R are found and used to find the rotational speed of the clutch output shaft based on the gear ratio of the differential. Then, the converted output shaft rotational speed Nsr of the clutch output shaft is calculated. This converted output shaft rotational speed Nsr is the rotational speed of the clutch output shaft converted to the rotational speed at the position of the output shaft of the electric motor 4. Then, the 4WD controller 8 proceeds to step S530. It is also acceptable to detect the rotational speed of the clutch output shaft directly and use the detected value as the rotational speed of the clutch output shaft.

Detailed Description Text (74):
In step S530, the 4WD controller 8 calculates the weighted average of the regularly acquired converted output shaft rotational speed Nsr and calculates the average rate of change DNsr of the converted output shaft rotational speed Nsr. Then, the 4WD controller 8 proceeds to step S540. The average rate of change DNsr is a value that corresponds to the rotational acceleration of the clutch output shaft.

Detailed Description Text (76):
Thus, the target rotational speed MNm is larger than the converted output shaft rotational speed Nsr of the clutch output shaft by a prescribed rotational speed difference Nmofs that is corrected by an average rate of change DNsr that corresponds to the rotational acceleration of the clutch output shaft multiplied by a constant value GDV.

Detailed Description Text (77):
The prescribed rotational speed difference Nmofs is a constant that is found through experimentation or the like, and that is set to a value that prevents shock caused by torque fluctuations when the clutch 12 connects from being transmitted to the passengers in the vehicle or suppresses such shock to such a degree that it does not bother the passengers. When the clutch 12 connects, there is backlash that needs to be eliminated between the clutch 12 and the subordinate drive wheels 3L and 3R. Consequently, torque fluctuations associated with connecting the clutch 12 are not recognized as shock so long as they are small. Furthermore, it is also acceptable to make the prescribed rotational speed difference Nmofs a variable that is, for example, inversely proportional to the rotational speed of the electric motor 4 or the rotational speed of the clutch output shaft.

Detailed Description Text (78):
The value GDV in the previous equation is a constant that serves to convert the

average rate of change DNsr into a value corresponding to the response delay of the clutch operation (i.e., a constant that serves to convert the average rate of change into the amount by which the converted output shaft rotational speed Nsr increases or decreases during the response delay period of the clutch 12). In other words, due to the response delay of the clutch operation, it is anticipated that the larger the average rate of change DNsr is, the larger will be the amount by which the rotational speed of the clutch output shaft exceeds the aforementioned detected value when the clutch 12 actually connects. This amount is corrected with the value DNsr.times.GDV.

Detailed Description Text (80):
The value of the prescribed correction amount DNm should be set in view of the control error amount. It is also acceptable to correct the prescribed correction amount DNm based on the rotational acceleration .DELTA.Nm (rate of change of the rotational speed) of the electric motor 4 (e.g., add an amount corresponding to .DELTA.Nm). In other words, the larger the rotational acceleration .DELTA.Nm of the electric motor 4 is, the larger the value of the prescribed correction amount DNm is set in anticipation of overshooting of the motor rotational speed Nm during the response delay of the clutch operation, thus achieving the same effect as reducing the value of the target rotational speed MNm. Since the effect is the same as directly correcting the target rotational speed MNm in accordance with the rotational acceleration .DELTA.Nm of the electric motor 4, it is also acceptable to correct the target rotational speed MNm in accordance with the rotational acceleration .DELTA.Nm of the electric motor 4 instead of varying the prescribed correction amount DNm in accordance with the rotational acceleration .DELTA.Nm of the electric motor 4. In this latter case, a value obtained by multiplying the rotational acceleration .DELTA.Nm of the electric motor 4 by an experimentally determined coefficient should be subtracted from the target rotational speed MNm. Furthermore, the same effect as is achieved by correcting the target rotational speed MNm directly using the average rate of change DNsr can also be achieved by changing the value of the prescribed correction amount DNm in accordance with the average rate of change DNsr.

Detailed Description Text (81):
In step S560, the clutch control section 8D connects the clutch 12. Then, the control loop ends and returns to the beginning.

Detailed Description Text (83):
In step S570, the 4WD controller 8 finds the throttle opening (acceleration instruction amount) based on the signal from the accelerator sensor 40 and determines if the throttle opening is greater than 5%. If the throttle opening is greater than 5%, then the 4WD controller 8 proceeds to step S580, where the backlash elimination flag GATAFLG is set to 0 and the control loop ends and returns to the beginning. If the throttle opening is determined to be less than or equal to 5% in step S570, then the 4WD controller 8 proceeds to step S520 and the clutch 12 is connected if it is not already connected.

Detailed Description Text (84):
Now the processing executed by the clutch connection determining section 8L will be described. In accordance with a prescribed sampling time cycle (e.g., 10 milliseconds), the clutch connection determining section 8L executes its processing through the following sections in order as shown in FIG. 10: a wheel stop estimating section, a motor stop estimating section, an external disturbance detecting section, a clutch connectability determining section, and a clutch connection flag outputting section.

Detailed Description Text (86):
In step S1010, the 4WD controller 8 determines if the average wheel speed V.sub.Wf of the rear wheels is larger than a minimum detectable rotational speed LWS, which corresponds to the lowest speed that can be detected with the detection precision

of the wheel speed sensors 27RL and 27RR. If it is determined that the average wheel speed V.sub.Wf is larger than the minimum detectable rotational speed LWS, then the 4WD controller 8 proceeds to step S1015. Meanwhile, if the average wheel speed V.sub.Wf is less than or equal to the minimum detectable rotational speed LWS, the 4WD controller 8 proceeds to step S1045. Step S1010 preferably constitutes a vehicle stop determining section that determines that the vehicle has stopped rotating if the output rotational speed value of the output part or shaft of the clutch 12 falls below the minimum detectable rotational speed LWS for the output shaft stop rotational speed sensor, e.g., the wheel speed sensors 27RL and 27RR.

Detailed Description Text (93):
In the above equation, the value TWD is the response delay time of the rear wheel speed sensors 27RL, 27RR and the 30 ms is the phase delay time caused by the computation executed in step S1020. In this embodiment, corrections are made for the response delay of the wheel speed sensors 27RL and 27RR and the phase delay of the aforementioned computation. It is also acceptable to correct for the response delay of the clutch operation.

Detailed Description Text (107):
In the above equation, the value TMD is the response delay time of the motor rotational speed sensor 26 and the 30 ms is the phase delay time caused by the computation executed in step S1120. In this embodiment, corrections are made for the response delay of the motor rotational speed sensor 26 and the phase delay of the aforementioned computation. It is also acceptable to correct for the response delay of the clutch operation.

Detailed Description Text (113):
Next, the processing executed by the external disturbance detecting section is explained based on FIG. 13. In this embodiment, the external disturbance detecting section executes processing to detect accelerator operations, brake operations, and other external disturbances that will change the stop time of the wheels.

Detailed Description Text (114):
First, in step S1210, the 4WD controller 8 finds the accelerator operation amount ACC based on the signal from the accelerator sensor 40 and then proceeds to step S1220.

Detailed Description Text (117):
In step S1240, the 4WD controller 8 determines if the accelerator was operated, i.e., if the accelerator operation amount ACC is larger than a prescribed value ACCTH (e.g., 5%). If it is larger, the 4WD controller 8 proceeds to step S1270. If it is not larger, the 4WD controller 8 proceeds to step S1250. The prescribed value ACCTH is a threshold value above which it is assumed that the traveling state of the vehicle is changing (experiencing external disturbance).

Detailed Description Text (121):
In step S1280, since and external disturbance exists, the 4WD controller 8 assigns the value 2 to the warning lamp status flag FWARN to request the warning lamp to flash and thereby inform the driver that the clutch 12 will not be connected during a temporary stop. Then, in step S1290, the 4WD controller 8 outputs the fact that the warning lamp status flag FWARN has a value of 2 or that the FWARN has changed to the warning lamp control section 8M and the control loop ends.

Detailed Description Text (123):
Next, the clutch connectability determining section will be described based on FIG. 14. Basically, FIG. 14 is a flow chart showing the processing sequence executed by the clutch connectability determining section for the vehicle driving force control apparatus illustrated in FIG. 1 of the illustrated embodiment of the present invention. The clutch connectability determining section includes a clutch stop determining section that is configured to determine that the clutch has stopped

rotating based on a determination of the occurrences of the detected output and input parameters, the first and second amounts of time estimated by the output and input shaft stop estimating sections.

Detailed Description Text (125):
In step S1420, the 4WD controller 8 determines if the required motor stopping time estimate value TMS is larger than the clutch response delay time CLD. If so, the 4WD controller 8 proceeds to step S1460. If the estimate value is less than or equal to the clutch response delay time CLD, then the 4WD controller 8 proceeds to step S1440.

Detailed Description Text (126):
Similarly, in step S1430, the 4WD controller 8 determines if the required wheel stopping time estimate value TWS is larger than the clutch response delay time CLD. If so, the 4WD controller 8 proceeds to step S1460. If the estimate value is less than or equal to the clutch response delay time CLD, then the 4WD controller 8 proceeds to step S1440.

Detailed Description Text (127):
In step S1440, the 4WD controller 8 determines if the value of the external disturbance detection flag FDS is 0, i.e., if the vehicle is free from external disturbances exceeding the tolerance. If so, the 4WD controller 8 proceeds to step S1450. Conversely, if the external disturbance detection flag FDS is not 0, the 4WD controller 8 proceeds to step S1460.

Detailed Description Text (128):
In step S1450, the 4WD controller 8 assigns the value 1 to the clutch connectability flag FCLP and proceeds to step S1470, where it sets the warning lamp status flag FWARN to 0, i.e., changes the warning light status information to off. Then, in step S1480, the 4WD controller 8 outputs the fact that the warning lamp status flag FWARN is 0 to the warning lamp control section before returning to the beginning of the control loop.

Detailed Description Text (129):
Next, the processing executed by the clutch control flag outputting section will be explained based on FIG. 15. First, the 4WD controller 8 determines if the value of the clutch connectability flag FCLP is 1. If the value of the clutch connectability flag FCLP is determined to be 1, then the 4WD controller 8 proceeds to step S1520 and outputs the clutch ON command to the clutch 12 via the clutch control section 8D.

Detailed Description Text (130):
Meanwhile, if the value of the clutch connectability flag FCLP is not 1, the clutch control section of the 4WD controller 8 returns to the beginning of the control loop, leaving the clutch connection status as is. In other words, if the clutch 12 is connected it is left connected, and if the clutch 12 is not connected it is left unconnected.

Detailed Description Text (133):
In step S610, the engine controller 18 computes the target output torque TeN requested by the driver based on the detection signal from the accelerator sensor 40 and then proceeds to step S620.

Detailed Description Text (142):
Now the operation of the vehicle driving force control apparatus constituted as described heretofore will be described.

Detailed Description Text (143):
The following explanation assumes the designated drive mode is set to the four-wheel drive mode. The clutch 12 is not connected when the designated drive mode is

set to the two-wheel drive mode.

Detailed Description Text (144):
When the torque transferred from the internal combustion engine 2 to the front wheels 1L and 1R is larger than the road surface reaction force limit torque, i.e., when acceleration slippage occurs in the front wheels 1L and 1R (which are the main drive wheels 1L and 1R), due to the road surface friction coefficient .mu. being small or the driver depressing the accelerator pedal 17 too deeply, the drive torque transferred to the front wheels 1L and 1R is controlled so as to approach the road surface reaction force limit torque of the front wheels 1L and 1R by having the generator 7 generate at a generator load torque Th corresponding to the magnitude of the acceleration slippage. As a result, acceleration slippage of the front wheels 1L and 1R (which are the main drive wheels) is suppressed.

Detailed Description Text (145):
Furthermore, the acceleration performance of the vehicle is improved because the surplus electric power generated by the generator 7 is used to drive the electric motor 4, which drives the rear wheels 3L and 3R (which are the subordinate drive wheels).

Detailed Description Text (147):
In a case where the rear wheels 3L and 3R are always driven, several energy conversions (mechanical energy.fwdarw.electrical energy.fwdarw.mechanical energy, etc.) take place and energy losses occur in accordance with the conversion efficiencies. Therefore, the acceleration performance of the vehicle declines in comparison with a case where only the front wheels 1L and 1R are driven. Consequently, it is preferred that driving of the rear wheels 3L and 3R be generally suppressed. Conversely, this embodiment takes into consideration the fact that when traveling on a slippery road surface or the like, even if all of the output torque Te of the engine 2 is transferred to the front wheels 1L and 1R, not all of the torque will be used as driving force. The driving force that cannot be utilised efficiently by the front wheels 1L and 1R is outputted to the rear wheels 3L and 3R and the acceleration performance is improved.

Detailed Description Text (148):
Additionally, in this embodiment, when the brake pedal 34 is released from a depressed condition in order to accelerate the vehicle while traveling, the 4WD controller 8 calculates a target voltage GaV for the electric motor 4 in accordance with a backlash elimination-purpose target motor torque GaTm that is proportional to the stroke speed of the brake pedal 34 when it is initially released. As a result, the electric motor 4 generates a very small torque while the clutch 12 is connected. The very small torque, which is not sufficient to drive the rear wheels 3L and 3R (subordinate drive wheels), acts on the torque transfer path between the electric motor 4 and the rear wheels 3L and 3R and functions to eliminate any backlash (play) existing among the mechanisms (i.e., the reduction gear 11, the clutch 12, and the differential gear 13) comprising the torque transfer path.

Detailed Description Text (149):
Afterwards, when acceleration slippage occurs at the front wheels 1L and 1R and the vehicle enters the four-wheel drive state, the backlash has already been eliminated as just described. Consequently, not only is the occurrence of shock resulting from backlash in the power transmission system prevented, but also the response of the rear wheels 3L and 3R to being driven by the electric motor 4 is improved because the backlash in the power transmission system has already been eliminated before the motor 4 starts driving the rear wheels 3L and 3R. In short, the response of the vehicle when it shifts into the four-wheel drive mode is improved. Then, when the motor torque for the four-wheel drive mode becomes larger than the small motor torque GaTh, the actual motor torque will be the motor torque for four-wheel drive mode.

Detailed Description Text (150):
If the speed of the brake pedal 34 is fast when it is initially released, it is highly likely that the accelerator pedal 17 will be depressed and acceleration will commence, i.e., the transition to the four-wheel drive mode will be executed early. In this embodiment, the faster the speed of the initially released brake pedal 34 is, the larger the value to which the very small torque GaTh of the motor 4 is set. As a result, the brake release speed is faster, backlash is eliminated earlier so that earlier transition to the four-wheel drive mode can be accommodated.

Detailed Description Text (151):
Even if the internal combustion engine 2 is in a driving state, it is not always necessary to eliminate the backlash, i.e., when the vehicle is in a non-driving range in which the output torque of the internal combustion engine 2 is not transmitted to the main drive wheels, i.e., the front wheels 1R and 1L. Therefore, the generation of unnecessary small torque is avoided in step S420 by preventing the execution of backlash elimination processing. In other words, wasting of electric energy is prevented by preventing the generation of a small electric current by the generator 7. Moreover, it is also acceptable to design the control program such that, even if backlash prevention is in progress, the 4WD controller 8 determines if the vehicle is in a driving range in step S540, for example, and, if the vehicle is in a non-driving range, the 4WD controller 8 proceeds to step S550 and stops the backlash elimination.

Detailed Description Text (152):
Backlash elimination is also stopped when the throttle opening exceeds a prescribed amount (e.g., 5% as in step S570). Although it is acceptable to set the prescribed throttle opening to 0%, in a case where acceleration slippage occurs and the vehicle shifts into the four-wheel drive mode slightly after the vehicle begins to move, it is possible that backlash will develop during the initial small amount of vehicle movement even though the backlash was eliminated in advance. Therefore, in the embodiment, the prescribed amount was set to 5%, which is the approximate throttle opening expected to exist when the accelerator pedal 17 has been depressed slightly and the vehicle either starts moving or starts to undergo acceleration slippage. Thus, it is generally preferred to set the prescribed throttle opening to an amount that approximates the throttle opening expected to exist when the accelerator pedal 17 has been depressed slightly and the vehicle either starts moving or starts to undergo acceleration slippage.

Detailed Description Text (153):
Moreover, when the clutch 12 is connected in order to execute backlash elimination, the clutch 12 is in such a state that the clutch input shaft (which is rotated by the electric motor 4) is rotating faster than the clutch output shaft by a prescribed rotational speed difference Nmofs. Thus, the backlash elimination can be completed early after the clutch 12 is connected and the acceleration is substantially free of a feeling of deficiency can be achieved when the vehicle shifts to a four-wheel drive mode.

Detailed Description Text (154):
This is in contrast with conventional control methods in which the clutch 12 is connected under such conditions that the rotational speed difference between the clutch input shaft and the clutch output shaft is zero. With such conventional control methods, the torque is increased gradually after the clutch is connected because the torque is substantially zero immediately after the clutch is connected. Consequently, strong acceleration of the subordinate drive wheels cannot be achieved and a feeling of speed loss occurs.

Detailed Description Text (155):
Moreover, with the present invention, although the clutch 12 experiences a response delay from when the clutch connection command is outputted until the clutch 12 actually connects, the difference between the rotational speed of the clutch output

shaft and the rotational speed of the clutch input shaft is kept within a prescribed range when the clutch 12 actually connects by correcting the target rotational speed MNm in accordance with the rotational acceleration of the clutch output shaft. As a result, the torque fluctuation that occurs when the clutch 12 is connected can be held within the targeted range without influencing the acceleration state of the vehicle, regardless of whether the vehicle is traveling at a very low rate of acceleration or a high rate of acceleration.

Detailed Description Text (156):
Also, if the rotational acceleration .DELTA.Nm of the electric motor 4 is large, the motor rotational speed Nm will be larger than the detected value when the clutch actually connects. However, similarly to the response delay just described, the rotational acceleration of the electric motor 4 can be prevented from exerting an adverse effect and the torque fluctuation that occurs when the clutch 12 is connected can be held within the targeted range by making corrections in accordance with the rotational acceleration .DELTA.Nm of the electric motor 4.

Detailed Description Text (157):
Since the processing just described in relation to connecting the clutch 12 is particularly effective when the vehicle proceeds immediately into the four-wheel drive mode after the backlash is eliminated, it is also acceptable to arrange for the processing to be executed when an accelerator switch turns ON, serving as a prediction of four-wheel drive mode. Another feasible option is to execute the processing of steps S520 to S550 (clutch connection timing control) only when the vehicle is traveling at or above a prescribed speed.

Detailed Description Text (158):
In the preceding explanations, the motor drive control was the same before and after connection of the clutch 12 during backlash elimination processing. However, it is also acceptable to use different motor drive control methods before and after the connection of the clutch. For example, before connecting the clutch 12, the electric motor 4 could be controlled by power control whereby the electric motor 4 is supplied with a fixed level of power (electric power). In such an arrangement, the motor torque would decrease as the rotational speed Nm of the electric motor 4 increased, thus making it possible to effectively suppress torque fluctuations when the clutch 12 is connected.

Detailed Description Text (159):
When the speed of the vehicle decreases to a very low speed corresponding to the minimum rotational speed that the wheel speed sensors 27RL and 27RR can accurately detect and it is estimated that the vehicle will stop, the driving force control apparatus computes the required motor stopping time estimate value TMS (which is an estimate of the time required for the rotation of the electric motor 4 to stop, i.e., for the input shaft of the clutch 12 to stop) and the required wheel stop time estimate value TWS (which is an estimate of the time required for the rotation of the rear wheels 3L and 3R to stop, i.e., for the output shaft of the clutch 12 to stop) using a deceleration value based on the detection results of the wheel speed sensors 27RL and 27RR up to the point in time when the very low speed was reached. Since the connection command is sent to the clutch 12 when the larger of the required motor stopping time estimate value TMS and the required wheel stopping time estimate value TWS has elapsed, the clutch 12 is connected under conditions where both the input shaft and output shaft of the clutch 12 have definitely stopped and the occurrence of shock when the clutch is connected can be suppressed.

Detailed Description Text (160):
When the times for the electric motor 4 and the rear wheels 3L and 3R to stop rotating are estimated, the values of the required motor stopping time estimate value TMS and the required wheel stopping time estimate value TWS are made more accurate by taking into account the response delay time of the sensors and the

phase delay of the computations. Furthermore, wasted time before connecting the clutch 12 is eliminated by taking into account the operation response delay of the clutch 12 and issuing the clutch connection command before the electric motor 4 or rear wheels 3L and 3R have actually stopped completely.

Detailed Description Text (161):
When the clutch connection is executed in this manner, the clutch 12 is already connected before the vehicle starts moving in cases where the vehicle starts moving from a stopped condition. Consequently, the vehicle has good response to the transition to the four-wheel drive and the vehicle can be provided with the required starting performance and acceleration performance.

Detailed Description Text (162):
Thus, when the vehicle is in the four-wheel drive mode, the clutch 12 is generally connected in advance when starting into motion from a stop, which is a time when acceleration slippage occurs easily. However, it is also acceptable to execute the previously described backlash elimination processing when starting into motion from a stop, thus preventing shock caused by backlash in the drive train when the vehicle is first starting into motion. Also, since the clutch 12 is connected in advance, it is not necessary to execute clutch connection processing.

Detailed Description Text (163):
When the vehicle decelerates, passes the very low speed, and stops briefly before starting into motion again, it might be assumed that in some cases the electric motor 4 will be rotating in an unloaded state during the brief stop of the vehicle. With this embodiment, in cases where the vehicle starts into motion before the rotation of the electric motor 4 stops, the clutch 12 is connected while the vehicle is stopped and the occurrence of shock when the clutch 12 is connected is avoided.

Detailed Description Text (164):
Another possible scenario is that after the vehicle reaches the very low speed and it is estimated that the vehicle will stop, the driver will depress the accelerator and request acceleration before the electric motor 4 and the wheels 3L and 3R stop rotating. In such a case, it is highly probable that the electric motor 4 or wheels 3L and 3R will not be in a stopped state when the estimated stopping time has elapsed because the traveling conditions of the vehicle will have changed and the required wheel stopping time estimate value TWS will no longer accurate. Therefore, in such a case, output of the clutch connection command is stopped and the occurrence of shock caused by connection of the clutch 12 is prevented.

Detailed Description Text (165):
Since, as just described, it is possible that the clutch 12 will not be connected when the vehicle stops, it is also acceptable to design the driving force control apparatus such that, when the vehicle starts moving, it determines if the clutch 12 is connected and, if the clutch 12 is not connected, executes backlash elimination control.

Detailed Description Text (166):
Similarly, another possible scenario is that after the vehicle reaches the very low speed and it is estimated that the vehicle will stop, the driver will operate the brake and change the braking force acting on the vehicle before the electric motor 4 and the wheels 3L and 3R stop rotating. In such a case, it is highly probable that the electric motor 4 or wheels 3L and 3R will not be in a stopped state when the estimated stopping time has elapsed because the traveling conditions of the vehicle will have changed and the required wheel stopping time estimate value TWS will no longer accurate. Therefore, in such a case, output of the clutch connection command is stopped and the occurrence of shock caused by connection of the clutch 12 is prevented. Moreover, it is also acceptable to execute this processing only in cases where the brake pedal 34 is operated in the brake release direction such that

the braking force becomes smaller. This is acceptable because if, conversely, the brake pedal 34 is depressed further and the braking force becomes larger, it can be estimated that the rotation will stop earlier than the stopping time estimate value, i.e., the rotation of the rear wheels 3L and 3R will already have stopped when the estimated stopping time finishes elapsing.

Detailed Description Text (167):
When the vehicle is in the two-wheel drive mode and the vehicle speed is below the very low speed, operation of the accelerator pedal 17 or brake pedal 34 will have no effect on the required motor stopping time estimate value TMS if the clutch 12 is in the disconnected state.

Detailed Description Text (168):
FIGS. 17 and 18 show example time charts of the clutch connection processing executed when the vehicle stops. As shown in FIG. 17, when the vehicle speed decreases, the driving force control apparatus finds the deceleration based on the detection values obtained from the wheel speed sensors 27RL and 27LL and the motor rotational speed sensor 26 up to the point when the respective minimum detectable rotational speeds LWS and LMS are reached. Based on the deceleration, the driving force control apparatus calculates the required wheel stopping time estimate value TWS and the required motor stopping time estimate value TMS at which the rotational speeds of the wheels and the electric motor 4, respectively, will become zero. Since both the required motor stopping time estimate value TMS and the required wheel stopping time estimate value TWS are counted down and the clutch connection command is issued when both values have reached zero, the clutch 12 can be connected under conditions where the rotations of both the input shaft and the output shaft of the clutch 12 have definitely stopped.

Detailed Description Text (169):
While the required wheel stopping time estimate value TWS is being counted down, connection of the clutch 12 is prohibited if the brake or accelerator is operated beyond certain tolerances. However, as shown in FIG. 18, connection of the clutch 12 is not prohibited when the brake pedal 34 is operated below the tolerance level and, after connection of the clutch 12 has been completed, the connected state of the clutch 12 achieved while the vehicle was stopped is maintained even if the accelerator pedal 17 is operated beyond the tolerance level.

Detailed Description Text (172):
As used herein, the following directional terms "forward, rearward, above, downward, vertical, horizontal, below and transverse" as well as any other similar directional terms refer to those directions of a vehicle equipped with the present invention. Accordingly, these terms, as utilized to describe the present invention should be interpreted relative to a vehicle equipped with the present invention.

Current US Original Classification (1):
701/67

CLAIMS:

1. A vehicle driving force control apparatus for a vehicle having a clutch installed in a torque transfer path from a drive source to a wheel, the clutch having an input part connected to the drive source and an output part connected to the wheel, the vehicle driving force control apparatus comprising: an output rotational speed sensor configured to detect an output rotational speed of the output part of the clutch and produce an output rotational speed value; an input rotational speed sensor configured to detect an input rotational speed of the input part of the clutch and produce an input rotational speed value; an output stop estimating section configured to estimate that rotation of the output part has stop rotating upon an occurrence of a detected first parameter that is based on the output rotational speed value received from the output rotational speed sensor; an

input stop estimating section configured to estimate that rotation of the input part has stop rotating upon an occurrence of a detected second parameter that is based on the input rotational speed value received from the input rotational speed sensor; a vehicle stop determining section configured to determine whether the vehicle has stopped; a clutch stop determining section configured to determine that the clutch has stopped rotating based on a determination of the occurrences of the detected output and input parameters, upon the vehicle stop determining section determining that the vehicle has stopped; and a clutch connection command outputting section configured to output a clutch connection command to connect the clutch, upon the clutch stop determining section determining that the clutch has stopped rotating.

2. The vehicle driving force control apparatus according to claim 1, wherein the output stop estimating section is further configured such that the detected first parameter is a first amount of time that is an estimation of time for the output part to stop rotating based on the output rotational speed value received from the output rotational speed sensor; the input stop estimating section is further configured such that the detected second parameter is a second amount of time that is an estimation of time for the input part to stop rotating based on the input rotational speed value received from the input rotational speed sensor; and the clutch stop determining section is further configured such that the clutch has been determined to have stopped rotating based on the first and second amounts of time estimated by the output and input stop estimating sections having elapsed, upon the vehicle stop determining section determining that the vehicle has stopped.

3. The vehicle driving force control apparatus according to claim 2, wherein the vehicle stop determining section further configured to determine that the vehicle has stopped if the output rotational speed value of the output part falls below a minimum detectable rotational speed for the output rotational speed sensor.

4. The vehicle driving force control apparatus according to claim 2, further comprising an acceleration instruction sensor configured to detect an acceleration instruction of the vehicle; and a clutch connection prohibiting section configured to prohibit the output of the clutch connection command by the clutch connection command outputting section, upon the acceleration instruction sensor detecting the acceleration instruction of the vehicle, regardless of the vehicle stop determining section determining whether the vehicle has stopped.

5. The vehicle driving force control apparatus according to claim 2, further comprising a brake operation amount sensor configured to detect a brake operation amount of the vehicle; a brake operation change amount determining section configured to determine whether a change in the brake operation amount per unit time is at least equal to a prescribed value, using the brake operation amount detected by the brake operation amount sensor; and a clutch connection prohibiting section configured to prohibit the clutch connection by the clutch connection command outputting section, upon the brake operation change amount determining section determining that the change in the brake operation amount per unit time is at least equal to the prescribed value, regardless of the vehicle stop determining section determining whether the vehicle has stopped.

6. The vehicle driving force control apparatus according to claim 2, wherein the clutch stop determining section is further configured to modify the first and second amounts of time estimated by the input and output stop estimating sections to take into account a response delay time of the clutch.

7. The vehicle driving force control apparatus according to claim 1, further comprising an acceleration slippage detection section configured to detect if acceleration slippage is occurring in a drive wheel that is driven by a vehicle drive source; and a generator control section configured to control a generation load torque of a generator to substantially correspond to an acceleration slippage

magnitude of the drive wheel, when the acceleration slippage detection section estimates acceleration slippage occurring in the drive wheel.

8. The vehicle driving force control apparatus according to claim 1, further comprising a drive mode selection section configured to select one of a multi-wheel drive mode in which at least the wheel driven by the drive source connected through the clutch is driven and at least one other drive wheel is driven by a drive source not connected by the clutch, and a non-all wheel drive mode in which at least the clutch disconnects the drive source connected to the wheel through the clutch, while the vehicle is traveling; and the clutch connection command outputting section being further configured to output the clutch connection command to connect the clutch when the multi-wheel drive mode has been designated.

9. A vehicle driving force control apparatus for a vehicle having a clutch installed in a torque transfer path from a drive source to a wheel, the clutch having an input part connected to the drive source and an output part connected to the wheel, the vehicle driving force control apparatus comprising: output rotational speed detecting means for detecting an output rotational speed of the output part of the clutch and produce an output rotational speed value; input rotational speed detecting means for detecting an input rotational speed of the input part of the clutch and produce an input rotational speed value; output stop estimating means for estimating that rotation of the output part has stop rotating upon an occurrence of a detected first parameter that is based on the output rotational speed value received from the output rotational speed detecting means; input stop estimating means for estimating that rotation of the input part has stop rotating upon an occurrence of a detected second parameter that is based on the input rotational speed value received from the input rotational speed detecting means; vehicle stop determining means for determining whether the vehicle has stopped; clutch stop determining means for determining that the clutch has stopped rotating based on a determination of the occurrences of the detected output and input parameters, upon the vehicle stop determining means determining that the vehicle has stopped; and clutch connection command outputting means for outputting a clutch connection command to connect the clutch, upon the clutch stop determining means determining that the clutch has stopped rotating.

10. The vehicle driving force control apparatus according to claim 9, wherein the output stop estimating means is further configured such that the detected first parameter is a first amount of time that is an estimation of time for the output part to stop rotating based on the output rotational speed value received from the output rotational speed detecting means; the input stop estimating means is further configured such that the detected second parameter is a second amount of time that is an estimation of time for the input part to stop rotating based on the input rotational speed value received from the input rotational speed detecting means; and the clutch stop determining section is further configured such that the clutch has been determined to have stopped rotating based on the first and second amounts of time estimated by the output and input stop estimating means having elapsed, upon the vehicle stop determining means determining that the vehicle has stopped.

11. A vehicle driving force controlling method for a vehicle having a clutch installed in a torque transfer path from a drive source to a wheel, the clutch having an input part connected to the drive source and an output part connected to the wheel, the method comprising: detecting an output rotational speed of the output part of the clutch and produce an output rotational speed value; detecting an input rotational speed of the input part of the clutch and produce an input rotational speed value; estimating that the rotation of the output part has stop rotating upon an occurrence of a detected first parameter that is based on the output rotational speed value; estimating that the rotation of the input part has stop rotating upon an occurrence of a detected second parameter that is based on the input rotational speed value; determining whether the vehicle has stopped; determining that the clutch has stopped rotating based on a determination of the

occurrences of the detected output and input parameters, upon determining that the vehicle has stopped; and outputting a clutch connection command to connect the clutch, upon determining that the clutch has stopped rotating.

12. The vehicle driving force controlling method according to claim 11, wherein the estimating of the output part has stopped rotating is conducted by estimating a first amount of time as the detected first parameter that is an estimate of the output part stopping rotation based on the output rotational speed value; the estimating of the input part has stopped rotating is conducted by estimating a second amount of time as the detected second parameter that is an estimate of the input part stopping rotation based on the input rotational speed value; and the determining that the clutch stopped rotating based on the first and second amounts of time having elapsed, upon determining that the vehicle has stopped.

13. A vehicle driving force control apparatus for a vehicle having at least one first drive wheel and at least one second drive wheel, the vehicle driving force control apparatus comprising: a first drive source configured to transmit a first drive torque the first drive wheel; a clutch installed in a torque transfer path formed between the first drive source and the first drive wheel, the clutch having an input part connected to the first drive source and an output part connected to the first drive wheel; an output rotational speed sensor configured to detect an output rotational speed of the output part of the clutch and produce an output rotational speed value; an input rotational speed sensor configured to detect an input rotational speed of the input part of the clutch and produce an input rotational speed value; an output stop estimating section configured to estimate that rotation of the output part has stop rotating upon an occurrence of a detected first parameter that is based on the output rotational speed value received from the output rotational speed sensor; an input stop estimating section configured to estimate that rotation of the input part has stop rotating upon an occurrence of a detected second parameter that is based on the input rotational speed value received from the input rotational speed sensor; a vehicle stop determining section configured to determine whether the vehicle has stopped; a clutch stop determining section configured to determine that the clutch has stopped rotating based on a determination of the occurrences of the detected output and input parameters, upon the vehicle stop determining section determining that the vehicle has stopped; and a clutch connection command outputting section configured to output a clutch connection command to connect the clutch, upon the clutch stop determining section determining that the clutch has stopped rotating.

14. The vehicle driving force control apparatus according to claim 13, wherein the output stop estimating section is further configured such that the detected first parameter is a first amount of time that is an estimation of time for the output part to stop rotating based on the output rotational speed value received from the output rotational speed sensor; the input stop estimating section is further configured such that the detected second parameter is a second amount of time that is an estimation of time for the input part to stop rotating based on the input rotational speed value received from the input rotational speed sensor; and the clutch stop determining section is further configured such that the clutch has been determined to have stopped rotating based on the first and second amounts of time estimated by the output and input stop estimating sections having elapsed, upon the vehicle stop determining section determining that the vehicle has stopped.

15. The vehicle driving force control apparatus according to claim 14, wherein the vehicle stop determining section further configured to determine that the vehicle has stopped if the output rotational speed value of the output part falls below a minimum detectable rotational speed for the output rotational speed sensor.

16. The vehicle driving force control apparatus according to claim 14, further comprising an acceleration instruction sensor configured to detect an acceleration instruction of the vehicle; and a clutch connection prohibiting section configured

to prohibit the output of the clutch connection command by the clutch connection command outputting section, upon the acceleration instruction sensor detecting the acceleration instruction of the vehicle, regardless of the vehicle stop determining section determining whether the vehicle has stopped.

17. The vehicle driving force control apparatus according to claim 14, further comprising a brake operation amount sensor configured to detect a brake operation amount of the vehicle; a brake operation change amount determining section configured to determine whether a change in the brake operation amount per unit time is at least equal to a prescribed value, using the brake operation amount detected by the brake operation amount sensor; and a clutch connection prohibiting section configured to prohibit the clutch connection by the clutch connection command outputting section, upon the brake operation change amount determining section determining that the change in the brake operation amount per unit time is at least equal to the prescribed value, regardless of the vehicle stop determining section determining whether the vehicle has stopped.

18. The vehicle driving force control apparatus according to claim 14, wherein the clutch stop determining section is further configured to modify the first and second amounts of time estimated by the input and output stop estimating sections to take into account a response delay time of the clutch.

19. The vehicle driving force control apparatus according to claim 13, further comprising an acceleration slippage detection section configured to detect if acceleration slippage is occurring in a second drive wheel that is driven by a second drive source; and a generator control section configured to control a generation load torque of a generator to substantially correspond to an acceleration slippage magnitude of the second drive wheel, when the acceleration slippage detection section estimates acceleration slippage occurring in the second drive wheel.

20. The vehicle driving force control apparatus according to claim 13, further comprising a drive mode selection section configured to select one of a multi-wheel drive mode in which at least the first wheel driven by the first drive source connected through the clutch is driven and at least one second drive wheel is driven by a second drive source not connected by the clutch, and a non-all wheel drive mode in which at least the clutch disconnects the drive source connected the first wheel through the clutch, while the vehicle is traveling; and the clutch connection command outputting section being further configured to output the clutch connection command to connect the clutch when the multi-wheel drive mode has been designated.

21. The vehicle driving force control apparatus according to claim 13, further comprising a second drive source configured to transmit a second drive torque to the second drive wheel and a torque to a generator that supplies electrical power to the first drive source.

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